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THE TREND OF FLYING BOAT DEVELOPMENT.

COMMANDER H. C. RICHARDSON (C. C.), U. S. N.

The subject of trends is a large one, but fortunately in its very terms it at once averts the danger of prophecy, so that in what follows the writer does not have to pose as a prophet but rather as an interested observer. Again, if trends had to be determined from the few existing advanced types the conclusions might easily be erroneous. It is only necessary to be reminded of the perfection of "Floradora" hats, or of automobiles of twenty years ago; no inference is intended that "fads" exist in flying boats, but the art is so new and the progress so rapid that present approximations to perfection might easily in a few years look just as sad as a "Floradora" hat.

Let us first consider what a flying boat is. The National Advisory Committee for Aeronautics' Definition is as follows:

"Seaplane—An airplane designed to rise from and alight on the water."

"Boat Seaplane (or Flying Boat)—A form of seaplane having for its central portion a hull or boat which provides flotation in addition to serving as a fuselage. It is often provided with auxiliary floats or pontoons for lateral support on the water."

We thus see that it is a "seaplane" of a particular arrangement. In its conception it was a boat fitted with wings and not wings fitted with a boat or boats. The difference was more real at first than at present, for until the flying boat "arrived" seaplanes were primarily land planes in which float gear was substituted, more or less awkwardly, for the conventional land gear and frequently with only a glimmer of the essential requirements of seaplanes, or of the effects of such substitution upon aerodynamic qualities. In fact, some of the early conceptions of "flying boats" had more of the idea of a glorified speed boat than a craft capable of flight in all its phases. However, both types of arrangements of seaplanes as float and boat type have had their influence on development so that to arrive at a true conception of the present trend we must consider the early forms and development of each type as they gradually improved in both seaworthy and airworthy qualities.

To give a complete historic resumé of the step by step development would be tedious and cumbersome.

As a result of the Wrights' first flight at Fort Myer, Va., in 1908, Lieutenant George Sweet, U. S. N., a naval observer, recognized the possibility of naval use and suggested the use of floats. In 1908 Curtiss had attempted flights from Lake Keuka with the *Loon,* fitted with two canoe-shaped floats. Suction of the water on this type and small reserve power made the effort only partially successful.

From the start in this country, seaplane development has been fostered by the United States Navy.

Curtiss's first successful flight was made at San Diego in 1911 with a plane fitted with wing section floats, much like those used by Fabre, but he promptly switched to the sled profile box section float upon which his later successes and development was founded. In this development Lieutenant T. G. Ellyson, U. S. N., and Captain Paul Beck, U. S. A., were intimately associated; practically living in bathing suits and even riding on the decks of the floats to determine their action. Tip floats were naturally evolved from the outrigger canoes of the Hawaiians and East Indians.

In the fall of 1911 Captain Chambers initiated Model Basin experiments with twin floats and "Forlanini" blades and Lieutenant William McEntee, Construction Corps, U. S. N., at the Model Basin designed a deep "V" bottom set of floats for the Wright plane. Their construction was largely based on a pair of single stepped "V" bottom floats with vent tubes designed by W. Starling Burgess of Marblehead, Mass.

Later in the fall, the writer, then on duty at the Philadelphia Navy Yard, was assigned temporary duty at the Model Basin to study float development. From that time the Model Basin has been used in naval development work, to test out existing types and experimental types and from the models to develop and construct full sized floats which are tried out in service. Now, no designs are undertaken without a Model Basin check on their performance. This simple method has aided importantly in development and in avoidance of unsatisfactory types.

CONFLICTING REQUIREMENTS.

Seaplanes must meet requirements of airworthiness and seaworthiness. Airworthiness demands that the float system must be—

(*a*) As small and as light as practicable.

(*b*) Have good streamline form.

(*c*) Have little effect on balance, both statically and aerodynamically.

Seaworthiness requires:—

(*a*) Stability afloat, moored, adrift or underway, taxi-ing or towing.

(*b*) A proper reserve of buoyancy and stability.

(*c*) Minimum water resistance.

(*d*) Maneuvering ability, up, down and across wind and in a tideway.

(*e*) Ruggedness to meet the high forces encountered at "take-off" and landing, or adrift in a seaway.

(*f*) Simple construction and maintenance to facilitate fabrication, inspection and repairs.

A careful examination of these requirements will at once indicate a conflict requiring a compromise. A few comments will make this still more evident.

Central floats require not less than 80 per cent reserve buoyancy.

Twin floats require not less than 90-100 per cent reserve buoyancy.

Boats as a rule inherently exceed these figures, but in doing so they usually combine the functions of float and fuselage. This excess is not chargeable in full part to the flotation system.

Lightness in any form of floats can only be approached while utilizing material to the best advantages to provide essential strength.

The ideal form would be purely streamlined, but this form is hopeless as to disengagement from the water and as to water resistance at high speeds, unless steps are added or use is made of hydrovane blades. However, these latter devices can be employed judiciously without offending the requirements of low resistance too seriously. It appears inadvisable to attempt to gain lift by giving the floats lifting profile for the aspect ratio is adverse; it appears of more interest to keep the forms otherwise clean and to derive lift from the far more efficient wings.

In all types of seaplanes, so far devised, the center of gravity is relatively lower than in the case of land planes, and because

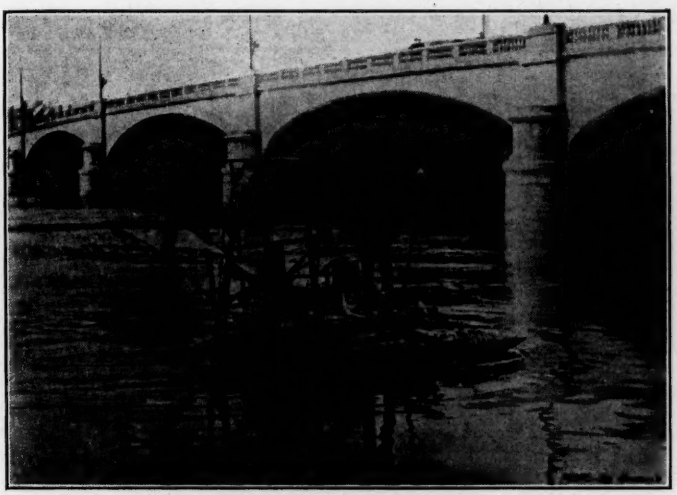


FIG. 1.—WRIGHT BROTHERS' CATAMARAN (1907).

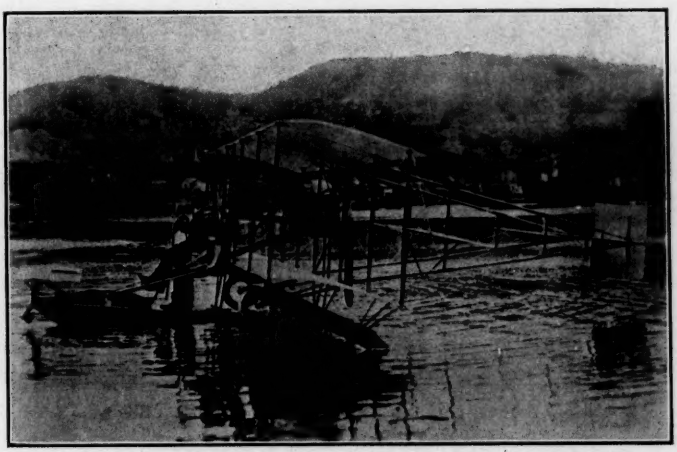


FIG. 2.—CURTISS' "TRIAD"—THE FIRST AMPHIBIAN (1911).

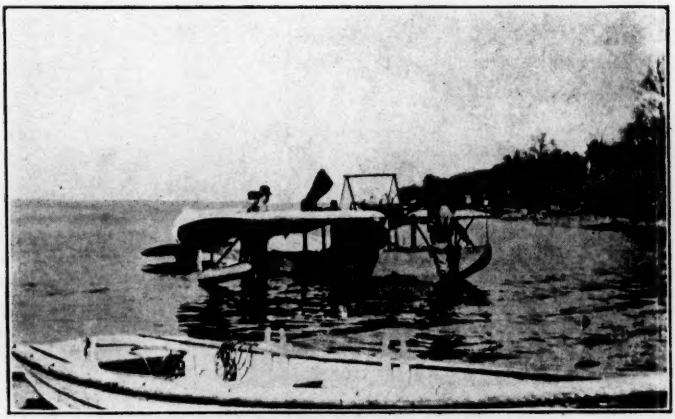


FIG. 3.—THE FIRST FLYING BOAT (1910).

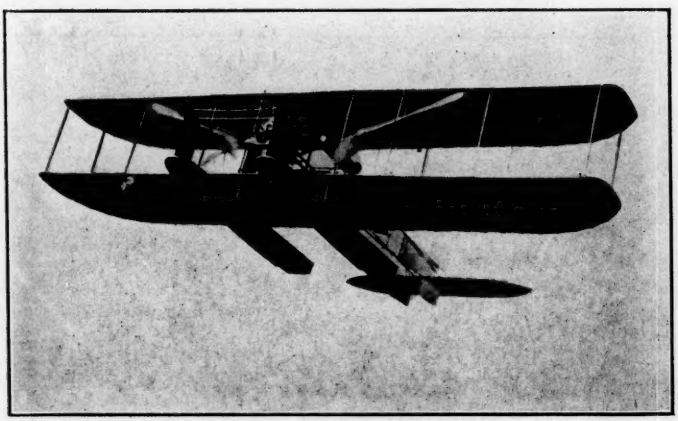


FIG. 4.—BURGESS TWIN FLOAT SEAPLANE (1911).

the engines require to be placed so the propellers are well clear of the water, there is an unavoidable separation of the center of gravity of large component weights. This has an effect on maneuverability because of the inertia involved about the axes of rotation.

Aerodynamically, the present-day arrangements are reasonably satisfactory and statically satisfactory balance is as readily, if not more readily, attained than in some land types.

Stability afloat is, of course, a prime essential and must remain so even when partial damage has occurred; this requires a reasonable subdivision into watertight compartments. It must be sufficient in strong winds affecting the non-furling surfaces. Adrift a seaplane should naturally weathercock into the wind, and yet be controllable and stable while several points off the wind. This quality is very useful in avoiding obstacles or delaying the drift towards a lee shore, or even as in the case of the *NC's* in the Trans-Atlantic flight and the *PN-9* in the Pacific flight, enabling actual courses to be steered within limits. Large boats and float planes must be capable of mooring out in high winds and rough seas without capsizing. Towing also presents a difficult problem for the floats are—relatively to other water craft—so short that strong yawing tendencies are present. Water rudders are only rarely employed.

Low water resistance is a problem in itself which is dealt with later on.

Surface maneuvering under power is called taxi-ing. The float arrangement and disposition of engines has a complex bearing on this problem. Single screw ships usually turn easily in one direction, but badly in the other, due to slip stream effects. Twin floats are more difficult to maneuver than single floats. Twin tractor or twin pusher or twin tandem types have a high degree of maneuverability when both units are available, but present special problems of seamanship if the power available is unsymmetrical.

Central tandem engines afford perhaps more control than single engine types by utilizing opposite slipstream effects available from the tractor and pusher propellers usually rotating in opposite senses.

In the early days of naval development it was considered that 45 miles per hour was about as high a speed as should be employed for the "take-off" of seaplanes, but already big flying boats have been handled in moderate seas at over 70 miles per hour and racing planes have gone higher than 90 miles per hour. When it is considered that regardless of how much the wings are lifting the pressure per unit area is a function of the angle at which the surface is presented and the square of the speed, the need for ruggedness will be appreciated. In the cases referred to, wooden bottoms of a half-inch or less in thickness have successfully withstood such service. A discussion as to how this has been accomplished belongs under the heading of "Structure" and will not be further discussed at this point. Adrift in a seaway, in a storm, it is more the sides of the floats that have to be considered, for the bottoms are, for reasons just cited, far more rugged than necessary for this requirement, and hogging and sagging stresses affect the hull trussing importantly. With twin floats, racking in and between floats is an additional problem of serious proportions, particularly underway in a cross chop or when crossing seas diagonally.

To meet the above requirements in a simple form of construction, having due regard to fabrication, inspection and ready repairs requires engineering ingenuity, experience and caution.

MODEL BASIN TESTS.

These tests are of great value, giving data on the following features:—

(*a*) The hull resistance at different initial trims and loadings, free to trim and at fixed trims.

(*b*) Trimming moments required underway.

(*c*) Righting moments at rest.

(*d*) Planing capacity.

(*e*) Porpoising tendency and location of spray.

In testing models of seaplanes, the models are towed at speeds corresponding to loads as affected by wing lift. The model is made to scale, usually one-quarter to one-twelfth the full size.

Certain assumptions are made as follows:—

(*a*) The ratio of speed of model to speed of full size is the square root of the length ratio; thus for a one-ninth size model the speeds for test are one-third those for the full size.

(*b*) The getaway speed of the full size is determined from the design; if, say it is 45 knots, it will be 15 knots for the one-ninth size model.

(*c*) It is assumed that the lift of the wings is proportional to the square of the speed and at getaway speed equals the weight of the plane; thus at one-half getaway, the wings lift one-quarter the weight and three-quarters is supported by the hull; at three-quarters getaway nine-sixteenths of the weight is supported by the wings and seven-sixteenths by the hull; at .9 getaway, .81 of the weight is borne by the wings, etc.

(*d*) At corresponding speeds for the model the resistance of the model is equal to the cube of the length ratio times that of the full size; thus for a one-ninth size model it is one-ninth cube ($1/9^3$) or one-seven hundred twenty-ninth (1/729) that of the full size.

(NOTE:—The above assumptions ignore certain factors):—

(*a*) The principal one which is; that the law of wing lift assumed implies that the wings act throughout at the angle of maximum lift, thus ignoring the effect on this angle of the change of trim of the hull. However, to allow for this would involve further assumptions more or less arbitrary. It should also be noted that up to .7 getaway the wing lift only accounts for one-half (.49) of the lift from the wings and at .3 getaway the vicinity of the hump resistance only calls for .09 of the weight carried by the wings so that at the hump the

error involved is small, whereas, above .7 getaway, the air controls are usually effective and the full available lift of the assumption is approximately available.

(*b*) In ship calculation, skin friction resistance is found to obey a different law than the "residual" wave and eddy-making resistance, the latter of which follows the law above assumed. Also, once planing speed, normally about .3 getaway, is attained, the amount of wetted surface subject to friction rapidly decreases so that it soon becomes a small factor. The errors involved in these factors have been found to be unimportant in interpreting the results of model tests. The assumptions being uniformly applied enable quite accurate determination of the relative merits of design.

Only the float models are towed, for the air eddies of the towing gear would introduce serious errors and would certainly greatly increase the cost of the model, which would further be exceedingly difficult to construct of the proper specific weight. (Among the early tests to check results, however, a complete scale model was used. The check was found satisfactory to a reasonable degree.)

The models are usually built of a solid block of wood. The model is towed from the dynamometer arm of the model basin carriage and the tow line pull and speed are automatically recorded on a chart for each run. To load the model properly it is counterweighted until just the right initial weight is water borne, according to the scale of the model; thus, a one-twelfth model of a 1728-pound plane would be loaded so that one pound rested on the water.

To compensate for wing lift, a small inclined flat blade of brass about 3 inches by 1 inch is submerged at a calibrated angle so that at the *model* getaway speed its down pull just equals the weight of the plane, e.g., in the above case, one pound.

The pull of the blade is readily applied to the counterweighted model and automatically applies a lift ranging with the square of the speed. Originally this was done by care-

fully counter-weighting for each speed run, but the new device is simpler and avoids the necessity of getting exactly the predetermining speed which is quite difficult to obtain with the model carriage. Models are usually run under two conditions and always towed about a point representing the line of thrust of the propeller. They are first set free to trim to the designed initial trim, say 3 degrees by the stern, up to at least the hump speed and sometimes higher, depending upon the speed at which "porpoising" occurs. This represents to a certain degree the behavior of the design when the water forces dominate the air forces available on the plane control surfaces.

Next, the model is run at one or more fixed trims, usually 6 degrees or 8 degrees, and the moments of the water forces acting are determined; the latter indicate the forces required in the air controls to maintain these attitudes. In special cases still further angles are tested.

"Porpoising" is the tendency of a model to jump and it is important this shall not be present in an increasing degree.

Although as indicated, the model basin gives important information, it has limitations. It can only indicate the behavior in a straight ahead run, and quantitatively only in smooth water, though by making small waves in the basin some idea of the result of running in rough water may be gained. These tests cannot simulate turning on the water or the results running down wind, though the effect of the latter could be simulated in special cases by reducing correspondingly the wing lift effect.

There are, however, many features of design which have been investigated which have importantly aided in the development of present designs by the introduction of some devices and the elimination of others.

DESIGN FEATURES.

(*a*) An easy entrance.
(*b*) Flat or hollow or "V" bottom.
(*c*) One or more steps; ventilated or non-ventilated.

(*d*) Outrigger tails on hulls vs. long tail hulls supporting the tail surfaces.

(*e*) Flush sides vs. so-called sponsons.

They avoid the following features:—

(*a*) Double ender, canoe or whaleboat lines.

(*b*) Multiple steps on or nearly on the same level.

(*c*) Strongly flared bows.

(*d*) Flat decks.

(*e*) Low freeboard.

(*f*) Downwardly curved lines aft of the first step.

(*g*) Twin hulls.

SUCTION.

Some of the early experiments at the model basin under the supervision of the writer taught what not to do, and not what to do. Consideration as to how to get the greatest buoyancy on the smallest float with the least possible wetted surface led to the design and test of a model with semi-circular section and shell pointed ends. This model tested at "corresponding" speeds gave remarkable results. At low speeds it was fine and the resistance was quite low. At one-half speed, with only three-fourths the initial weight, its resistance was not good, and the model indicated a failure to plane; at three-fourths speed, with only seven-sixteenths the initial load on the water, the model instead of rising proceeded to settle deeper underway and began to lift sheets of spray at each side throughout its length. At getaway speed, with no load water borne, but starting just in contact with the surface, this model was sucked down until the deck was flush with surface of the basin and lifted solid sheets of spray several feet above the surface; its resistance was very high. Analysis of the behavior showed that the downwardly curved lines were subjected to suction because of the type of water flow involved. (NOTE:—The reader may readily verify this powerful effect by a simple experiment, as follows:—Take a teaspoon or a tablespoon and suspend it vertically, bowl down with the handle between the

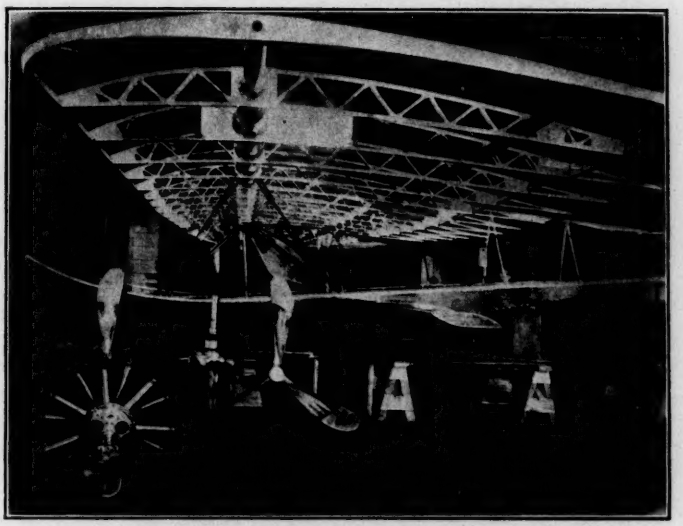


FIG. 5.—GALLAUDET STEPPED BOAT (1914).

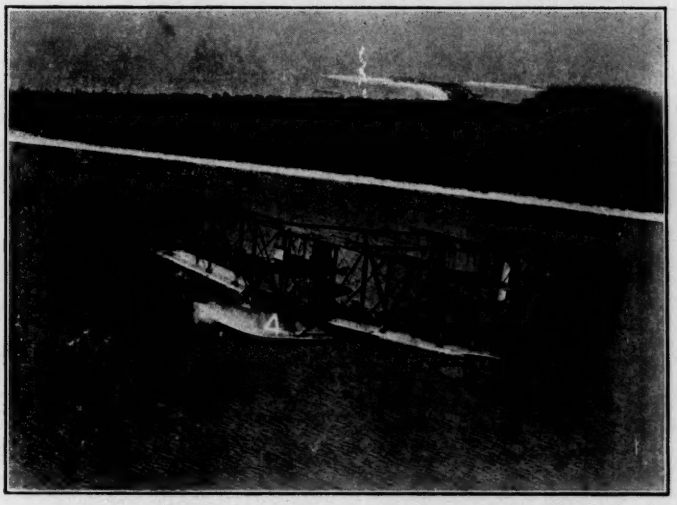


FIG. 6.—"NC-4" FIRST PLANE TO CROSS ATLANTIC (1919).

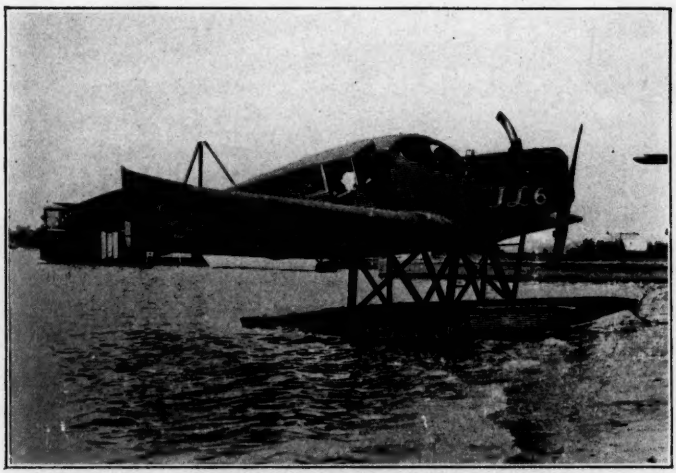


FIG. 7.—JUNKERS' ALL METAL PLANE (1919).

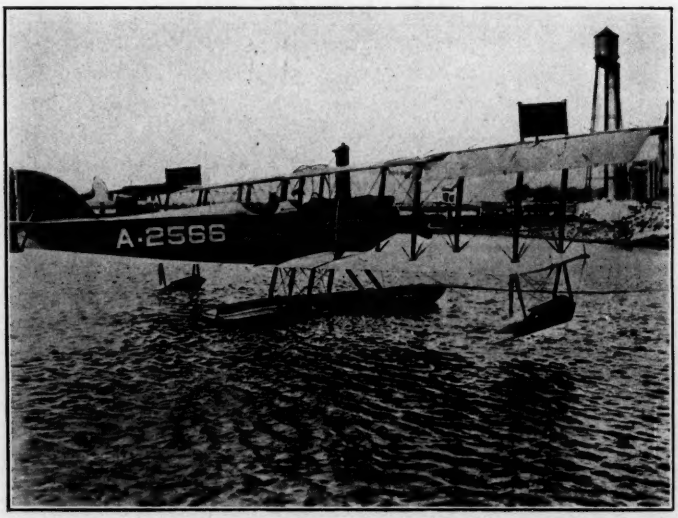


FIG. 8.—"N-9" PLANE WITH METAL FLOAT (1921)

thumb and forefinger. Now let the bottom of the bowl approach the stream from a faucet. As soon as the spoon touches the stream it will be found that the spoon is drawn into the stream with considerable force and the stream will be deflected away from the vertical.)

This phenomena of the production of suction or negative pressure on a curved surface by the flow of a stream of air or water over same, is the very essence of the success of the use of curved airfoils. When it is recalled that water is about 800 times as dense as air, it will be appreciated why it is necessary to avoid downwardly curving lines in the parts of hulls exposed to contact at high speed. Due to this effect, the double-ended "V" bottom floats designed at the model basin for the Wright airplane failed to "take off" with more than one passenger except in a strong head wind. The writer also saw a flying boat which, with one person aboard, could get the step a foot clear of the water, but which, due to suction on the curved bottom abaft the step, could not free its tail from the grip of the water. After this curvature was removed the same boat took the air easily with two passengers. More recently, a successful Austrian flying boat obtained through reparation was tested at Philadelphia. This hull had long straight lines from the step to the stern, but at the stern had strong up curvature. Accidentally the first effort to "take off" was successful, but on a second attempt, after planing was attained, the stern was permitted to again make contact. Then, in spite of every effort and even though the step of the float came clear of the surface, the tail could not be released from the water. On a third attempt, realizing this, a very quick getaway was attained by keeping the tail of the float well up as planing speed was approached.

This effect, in another form, is found when the sides of a float at the stern are vertical with strong curvature in plan form. An early experimental float had such a stern, the model tests were quite satisfactory, and a full size float with a duralumin skin was built accordingly. It was tried out on a Wright

plane at Annapolis. The first "take-off" and landing were successful, but on a second attempt, while just at planing speed, an effort to change course was made by rolling down the left wing. Much to the surprise of the pilot the plane suddenly whipped to the right. This was caused by suction on the left side of the stern well behind the center of gravity of the plane. The action was greater on the left side, because due to lowering the left wing the left side of the float was submerged more than the right. These forces were present, but balanced on the first flight. Later, similar effects were found with streamline triple floats and with a pair of floats derived from splitting a single float in the center line plane and by separating the parts, making a twin float.

Both the triple float and the twin float had a serious tendency to ground loop, which was cured by utilizing "side steps" to break up the suction effect.

These effects of suction are symmetrical under the conditions of the model basin tests. They are now well recognized and avoided.

Suction may occur under certain circumstances with even more serious consequences. Practically all seaplanes have curvature in the bottom forward of the main step and running to the bow. In general, the bottom is fairly flat just forward of the step, but from thereon curves and rises rapidly. In normal flight the bottom of the float is about level, but at high speed the bow is usually depressed. Under this condition it is possible for the curved portion under the bow to make the first contact with the water. It has happened in a number of instances that such contact has been made at high speed. The suction effect has been sufficient to cause failure of the bottom by bursting "outward," not inward, and the combined effect of suction under the bow and the rush of water into the damaged hull has been to cause the plane to somersault, usually tossing the crew clear and well ahead of the plane, even when they have been strapped in. For seaworthiness this disposition

of the bottom can hardly be eliminated, but knowledge of this effect should prevent its coming into action accidentally. In normal "take-offs" and landings it does not come into play.

One more instance of suction will about cover the important effects of this factor. Streamline tip floats have been used in which curvature of the bottom has been presented to the water flow, and this has only been permissible by the use of hydrovane blades projecting from the bottom to overcome this suction and to provide necessary planing properties. In one instance even these blades were insufficient under certain conditions and the wing of a large flying boat was swamped and nearly wrecked because of suction on the bottom of such a tip float.

Enough now appears to have been said on this subject to show the importance of avoiding this factor in the design of seaplanes.

STEPS.

The use of steps in the bottoms of fast *hydroplanes* has long been known. Apparently the possibility was first demonstrated by Dr. Rassmussen. The work of Henry Fauber on this line has been recognized, and he is probably the first to reduce designs to a practical form in his French experiments in which he employed multiple steps, and hollow V's.

The effect of a step is interesting. By its use, as the hull is driven faster and faster, the water flowing over the bottom does not follow the step but rises as an inverted water fall, from the step to the bottom next aft of it, at a greater distance away as the speed is increased. The inclination of the step also produces dynamic lift so that the hull rises more and more as speed is increased and finally only a small portion of the step is in action, the net effect is to greatly reduce the wetted surface, thus reducing skin friction and the planing effect eliminates hull resistance of the usual type due to displacement.

With hydroplane boats, stability on the water is of great

importance and the relation and number of steps must be given consideration with this in view.

With seaplanes the steps are used somewhat differently. They are still used in the first stage of planing to reduce skin friction but the bottom to the rear of the first step is turned up more sharply than usual in hydroplanes, to permit changing longitudinal trim; the bending would otherwise produce suction effects, and the steps help to break these too. There is a further important difference which is, that as the speed increases the air controls and the air lift become more effective, and, in fact, stability is no longer a requirement, but is to a degree a detriment, except for operation from relatively smooth water.

As above related, side steps have been used to break the effects of suction on the sides of the floats.

The usual form of step is in a plane perpendicular to the keel, but in a few cases, steps parallel to the keel have been used to provide a narrower bottom at the higher speeds.

From one to six or more steps have been employed in hydroplanes, but it is seldom that more than two steps are now used in seaplanes. The usual step in seaplanes is about three inches deep and parallel from keel to chine. The forward step is usually in the vicinity of a position directly under the center of gravity so that on the "take-off" the forces acting are well centralized. Poor location of this step will result in bad porpoising tendencies. Vent tubes have been found useful to break the suction behind the steps, where the chines are well covered; but in the more prevalent types of "V" bottoms this appears unnecessary.

"VEE" BOTTOMS.

The prime reason for the "V" bottom is to reduce the shock of landing or when planing at high speed; it further permits a design to conform more closely to streamline requirements and in general eliminates the curvature and suction prevalent with the conventional streamline hull. Flat bottoms, or even inverted "V" bottoms or hollow bottoms, are still popular with

certain designers and in smooth water they appear to get away a bit faster; but their employment is confined to small seaplanes and involves severe punishment on choppy water. Headed into the wind, too, they are clean so far as the crew being subjected to spray is concerned, but this is practically true also of the "V" bottom. From a resistance point of view, even steep "V's" up to 30 degrees on each side appear to plane nearly as well as flat bottoms and to require no additional power.

The straight "V" has one property which is disadvantageous and this is, that in its pure form besides the usual bow wave, there is carried with it a heavy sheet of spray which is dangerous to the propellers and is annoying to the crew in a cross wind. However, by making the "V's" hollow, and in particular by the use of sharp curvatures at the chines, either by the use of spray steps or built in form, it is possible to hold this spray down to the degree required. Hollowness does not appear of particular advantage at the step, but it is of considerable value at the bow to form good cruising lines. A too pronounced hollow "V" presents too much dead wood and causes tilting and wrenching on a skidding landing; if too pronounced forward it also affects steering.

A well-formed hollow bow turns the water down in an impressive manner and also keeps the bow from plowing too deep into head seas.

HYDROVANES OR BLADES.

The use of submerged blades to overcome the resistance and suction effects of conventional streamline hulls probably antedates the practical use of steps. Crocco and Ricaldini and Forlanini in Italy made classic experiments on these lines. In this country on July 29, 1897, at Chicago, Ill., L. E. Meacham succeeded in towing a boat fitted with blades at such a speed that the hull rose clear of the surface. The writer did the same thing with a canoe at Hampton, Va., in 1908, and later at Philadelphia in the summer of 1911 towed a dinghy fitted with warping blades for lateral control and a rudder and ele-

vator combined, by which the hull came clear of the surface. The elevator and warping were controlled by a stick and the rudder by a foot bar.

In 1907 the Wrights experimented with a catamaran supported by two blades driven by a twenty-horsepower airplane power plant taken from one of their planes.

Alexander Graham Bell and his engineer, Casey Baldwin, experimented with an elaboration of this scheme, and in 1919 attained a speed of 70 miles per hour using a Zeppelin-shaped hull mounted on a Venetian blind arrangement of blades. The blades were arranged in V form to provide differential action and to avoid punishment. At twelve miles an hour the hull cleared the water. Two Liberty engines and air propellers were used to drive this boat. It had excellent stability.

As a result of several experiments in 1911, blades were also used on seaplanes at San Diego in 1912, but were found to be tricky, though efficient.

Guidoni in Italy successfully employed them on seaplanes. Difficulty is encountered unless the "V" form is used, as flat blades when heavily loaded suddenly lose their lift as the surface is approached, as there is insufficient fluid following the top curvature of the blades. In fact, the stream over the back of the blade under these conditions suddenly splits off at about 60 degrees to the face of the blade, and this condition maintains itself until the blade sinks deeper.

In an early Curtiss experiment at San Diego in 1911, such blades were used to keep the nose of the main float from submerging at low speed and high thrust. Curtiss also used such a blade at his front wheel on his Albany-New York flight to prevent nosing over in case of a landing in the river.

FLOAT DISPOSITIONS.

The principal float dispositions in use today are the central float and the twin float arrangements.

In the central float arrangement the principal buoyancy is obtained in a central float. When this is attached to a fuselage

we have the familiar type of float seaplane and when the fuselage is combined in the float structure we have the boat seaplane. In both of these cases lateral stability depends upon wing tip floats, as the main hull is too narrow to provide the necessary righting moment.

In the twin float types a catamaran arrangement is made and the floats are separate from the fuselage. If the floats are short a tail float is frequently used to prevent turning over backward, but in this country the tails of the floats are usually long enough to make this unnecessary.

With the central float it is possible to bring the tip floats in from the wing tips and by increasing their buoyancy to obtain lateral stability and in some cases increased planing at low speeds; such an arrangement is termed triple float.

The use of twin floats is almost essential for short span planes, and even in larger planes designed for bombing and torpedo work.

A controversy as to the merits of twin vs. central floats has been waging since seaplanes came into existence and is still running strong. The writer from the start has stoutly maintained the advantages of the central float type and believes its advantages are now being generally recognized as being more seaworthy and maneuverable. This becomes increasingly apparent as the size increases as the racking forces between large twin floats become excessive. Briefly, it is easier structurally to carry a load over a pier than it is in the middle of a span. The central float also affords excellent protection for the propeller. It is apparent that it is easier to get the necessary displacement in one large hull than in two smaller hulls, and the weight and air resistance are less.

FLOAT CONSTRUCTION.

Until recently float construction has been almost entirely of wood in various forms, but from that it must not be inferred that metal has not been attempted. Until recently this has been due to lack of suitable material and skill in its use.

In the first box type floats the framing was simple comprising longitudinal center lines, keel and deck members, chine and keel stringers, numerous cross frames and a few bulkheads. The deck, sides and bottom were straight thin slabs. False keels took the wear and local reinforcements provided for strut and wire attachments.

With the advent of yacht builders the framing became more complex, but more efficient. Even as early as 1911 aluminum floats were built, and in 1913 a dural skin was used. But as floats became boats, the two-ply bottom came into existence and intermediate longitudinals were introduced to reduce the number of frames and floors required. At this stage the deck was usually a single ply over numerous frames, and watertightness of the deck was secured by cloth glued on. The bottoms were so thin that the usual calking was impracticable so that varnish or glued cloth was used between the bottom plies to obtain watertightness.

As hulls became larger, stick and wire or built-up trusses were used to furnish longitudinal strength, particularly where the boat type was given a long tail to support the tail surfaces. In floats the decks were often two-ply diagonal with cloth and glue; but in both floats and hulls, veneer decks and even veneer bottoms were used.

However, with wood construction soakage was almost inevitable and metal has been looked forward to. With the advent of reliable aluminum alloys this material is now generally accepted as the coming type, and with the duralumin skin comes naturally the duralumin framing.

But it "ain't all honey and it ain't all jam." Design difficulties are many, mostly from the cost and maintenance point of view and also from the corrosion point of view. But these difficulties are not insurmountable and are even now rapidly yielding to experience.

Having discussed the principal factors affecting design and construction, let us now review rapidly a few of the early efforts which have made the "Flying Boat" a reality. The

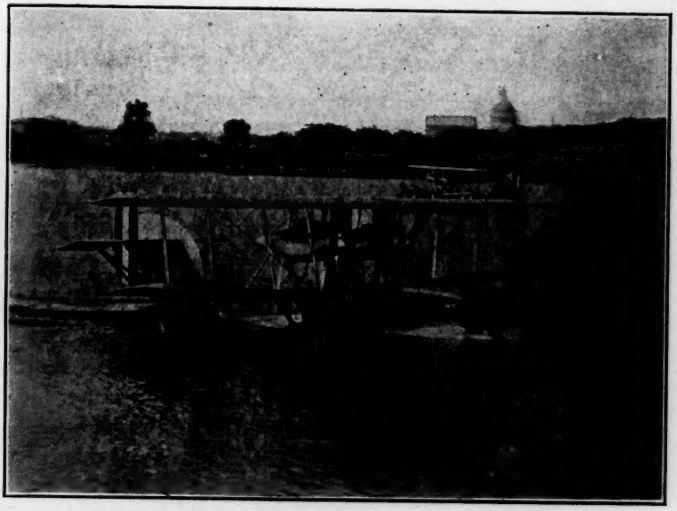


FIG. 9.—"PN-7" PREDECESSOR OF TYPE "PN-9."

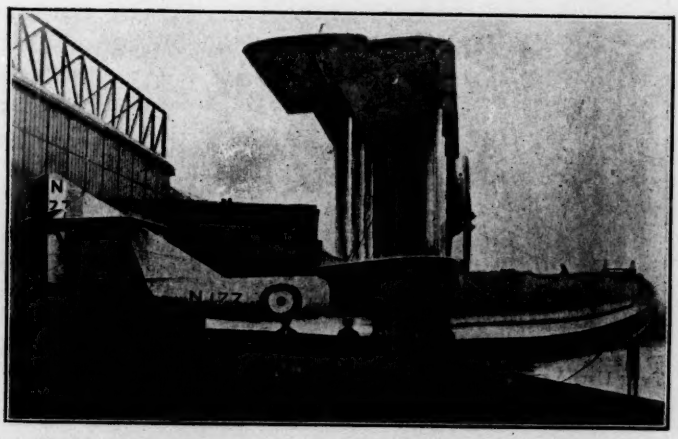


FIG. 10.—SHORT BROTHERS' METAL HULL FLYING BOAT.—(ENGLAND).

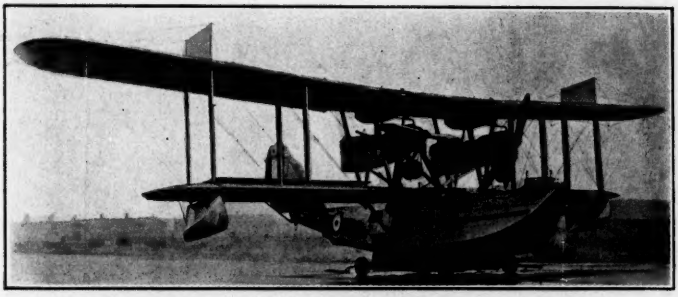


FIG. 11.—KINGSTON METAL HULL FLYING BOAT.—(ENGLAND).

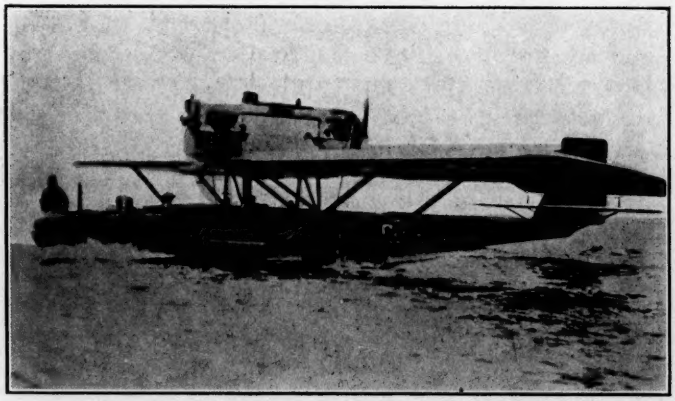


FIG. 12.—DORNIER WAL FLYING BOAT.—(ITALY).
TYPE USED BY FRANCO ON RECENT FLIGHT FROM SPAIN TO SOUTH AMERICA.

writer has endeavored to obtain an accurate chronology and believes the dates given are substantially correct, but apologizes for errors which may exist.

Apparently the first efforts at flight from the water date back to 1907, the year in which Bleriot and Archdeacon both succeeded in towing twin float gliders into the air. About the same time Kress, in Germany, succeeded in getting off the water, but the landing was not successfully negotiated. It was this year, too, as previously related, that the Wright brothers tried their catamaran fitted with blades and propelled by an aircraft power plant, but without wings (see Figure 1). It was about this time that Guidoni made successful experiments employing hydrovane blades.

In 1908-1909 the Aerial Experiment Association at Hammondsport, N. Y., achieved indifferent success with the *Loon* fitted with twin canoe-shaped floats and hydrovanes. In 1909 Fabre with three wing section floats succeeded in flight to and from the water.

Early in 1911 Curtiss, using one small wing section float in front and a larger wing section main float, plus a small hydrovane in front, and paddles and tire tubes for tip floats, succeeded in flight from and to the water at San Diego. Immediately following this success he changed to the sled type central float, which was quickly followed by the *Triad,* the first amphibian. (See Figure 2.)

Late in 1910, what appears to have been the first flying boat was designed and built by Loening for the Queen Company. However, flight was not attained until March, 1911. This boat had a step (Figure 3) slightly to the rear of the center of gravity. This same year Burgess developed twin floats, with steps and vent tubes (Figure 4). Alger, this year, built simple box floats of aluminum for the Coffin-Wright plane.

Early in 1912, Curtiss developed the flying boat, first with a knuckle, then with a wedge step at the knuckle, then a step under the center of gravity, and finally vent tubes. By the

fall of 1912 he delivered the first of a long series of successful flying boats to the Navy.

In rapid succession now came the Benoist, 1912; Columbia boat, 1913, and the Burgess in 1913 in this country, the Donnet-Leveque in 1913 in France, and the Macchi in Italy.

In 1914 Gallaudet brought out a monoplane stepped boat (Figure 5).

The big flying boat started with the *America* in 1914. This was the predecessor of the *H-12, H-16, HS* and *F5-L* flying boats used by England and the United States during the World War.

The war brought out the Macchi in Italy, the Levet le Pen and Donnet Denhaut and Tellier in France and the Austrian "K" boats. Germany stuck to twin floats, of which the Brandenburg seaplanes were the most famous.

During the war the U. S. Navy brought out the *NC-1* with three engines, at that time the largest flying boat in the world. At the same time the English brought out the *Porte Baby* and finally the *Felixstowe Fury,* the latter intended for a Trans-Atlantic flight.

Immediately after the war the *NC-1* was improved by adding another engine, resulting in the *NC-4,* the first plane to fly across the Atlantic (Figure 6).

From now on the metal development was rapid. In 1919 the Junkers all-metal plane was fitted with twin metal floats (Figure 7).

In 1921 the Navy started experiments in metal floats covering the *N-9* series at the Naval Aircraft Factory (see Figure 8), the Curtiss *TS* in 1922, the Brewster *TS* in 1922, and the Elias *EM* floats in 1924, and the Martin *MS* in 1923.

In 1922 the first American metal flying boat was developed by the Aeromarine Plane and Motor Co.; it was quite successful.

In 1922 the *Dornier Dolphin,* all-metal flying boat was developed by German engineers. In the same year the *PN-7* metal hull was developed at the Naval Aircraft Factory, Phila-

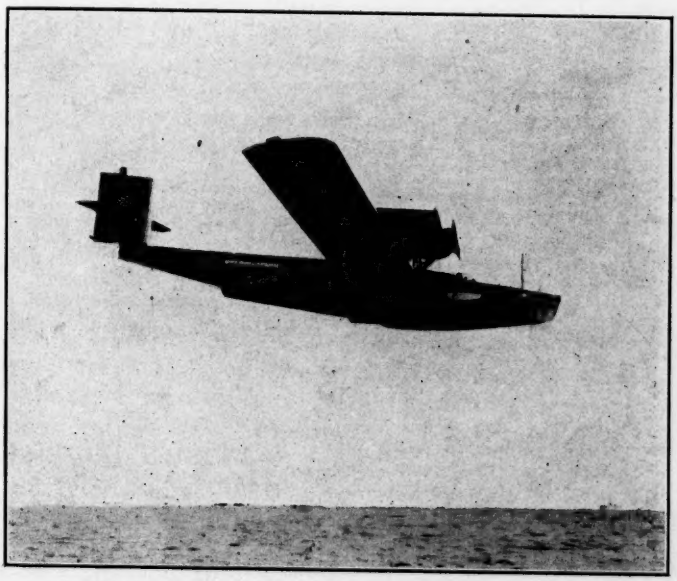


FIG. 13.—ROHRBACH FLYING BOAT.—(DENMARK).

delphia Navy Yard, which was the basis of the design of the *PN-9's* which took part in the Pacific flight of 1925 (Figure 9).

Contemporaneous development has brought out the Short (Figure 10) and Kingston (Figure 11) metal hulls in England, the Dornier Wal (Figure 12) in Italy, by German engineers and the Rohrbach (Figure 13) in Denmark.

At this writing the outstanding types are thus the *PN-9*, Short, Kingston, Dornier Wal and Rohrbach, the latter two being monoplanes, the Dornier Wal using sponsons to provide lateral stability, the Rohrbach using triple floats developed in this country before the war. The others use conventional tip floats. Both the "Wal" and the Rohrbach use tandem engines.

THE TREND.

From consideration of the factors and influences already referred to, not to mention many little side lights not herein attempted to enumerate, it appears fairly safe to conclude there is a trend and that it has certain definite features.

Foremost comes size; it is unsafe to put limits on this. At present the limits due to demand and cost appear to be well within practicable size, so it appears safe to say the trend is toward increase in size, and that means up to and beyond 30,000 pounds.

The advantages of size lies in increased seaworthiness, airworthiness, reliability and earning capacity.

Seaplanes have a great advantage over land planes. Their airdromes are almost everywhere, in every harbor, river, bay and sound. They compete principally with boat lines so the advantage of speed is greater than in competition with express trains, and in general they can approach closer to the heart of a city than any reasonable sized landing field can be economically located, in the larger cities.

Multiple engines will be used because single engines of the required capacity will only exist for the smaller planes; and multiple engines mean increased reliability and safety.

Air-cooled engines are at least equal competitors in certain sizes today and may become even more generally used. Metal propellers, steel or alloy, will displace wood.

In larger planes the central float is almost inevitable and the triple float arrangement will probably dominate. Sponson types are of inferior seaworthiness.

In seagoing planes, the monoplane will probably dominate as the wings are less subject to damage in heavy seas, in case of a forced landing. For inland waters, the biplane may dominate and in such cases tip floats may be retained.

Metal covered wings will only come slowly as the skin is disposed so that its weight is utilized to develop strength. Maintenance problems will also keep the cloth covered wing alive for years to come. Metal framed wings will soon be common.

The hull will carry the empennage (tail surfaces). It will have hollow "V" lines at the bow, to form a good cruising bow, with good freeboard. It will have not more than two steps.

The hull will be framed and plated with metal; alloy steel or aluminum alloy. It will be well subdivided and may incorporate double bottoms.

The fuel will gradually change from gasoline to heavy oil.

In the larger sizes, auxiliary engines may be employed for the handling of anchors and lines, drainage, lighting, signaling and propulsion afloat.

On passenger planes, life rafts will be carried.

Cruising speed will be in the neighborhood of 100 miles per hour.

Landing speed will be under 50 miles per hour and maximum speed above 130 miles per hour. Cruising will be possible with one engine cut out.

Pairs of engines will be frequently employed in tandem.

High lift wings, with some form of flap to reduce landing speed, will be employed.

On larger planes power will be used to operate the controls which also may be stabilized.

Navigating equipment will include turn indicators, induction compasses and the usual position finding equipment.

The prime requisite to development is demand, and demand rests upon dependability, which in turn means regularity and reliability.

PRACTICAL NAVAL APPLICATIONS OF THE THEORY OF LEAST SQUARES.

By COMMANDER KARL F. SMITH, U. S. N., MEMBER.

To most of us, a statement that one of the many problems which confront us from time to time can be solved, only by an application of the Theory of Least Squares, is enough to make us pass up the solution without a trial.

This perhaps is due to the fact that the teaching of this theory is often left until we have finished "The Calculus" instead of teaching it in high school where it should properly have its place, for whether we realize it or not, we use a part of this theory nearly every day of our lives. In my day, the theory was not even taught at the Naval Academy.

Just how difficult is this theory of Least Squares?

It is extremely simple, and wittingly or no, you and I have used it either correctly or incorrectly hundreds of times.

Suppose in making a rapid hydrographic survey you wished to establish the length of a certain base line between two signals.

You would probably take a steel tape, stretch it to a definite tension, and if you were a glutton for punishment, note the temperature and note the readings. Then you would correct these readings for sag of the tape and expansion. Undoubtedly you would repeat the procedure, and, strange to say, get a different result, for it is psychologically impossible—irrespective of the care taken, to obtain *absolutely* the same results, unless by accident. If you have no working knowledge of Least Squares these two different readings will undoubtedly force you for accuracy's sake to take a third, which will probably be different from either. Finally in desperation or disgust you might say—well, I'll average the readings and let it go at that!

Right here you have solved, and solved correctly, the length of the line, and in doing so have used this "Terrible Theory of Least Squares."

You probably don't know just how much error you have made, or how far out you may be, that is to say, your "limit of errors," but this, too, is easily determinable as will soon be seen.

The averaging of our recorded measurements as above is in accordance with a statement of the Theory of Least Squares as follows:

(1) GIVEN ANY NUMBER OF EQUALLY CAREFUL INDEPENDENT OBSERVATIONS OF ANY CONSTANT PHYSICAL QUANTITY (IN THIS CASE A FIXED DISTANCE), THE MOST PROBABLE VALUE IS THE ARITHMETICAL MEAN OF ALL OBSERVATIONS.

Now there is a very peculiar fact to be noted about an average value, or arithmetical mean as we have called it above. Irrespective of the number of observations averaged, if we take differences between each observation and the arithmetical mean, this difference being called a residual, and then square and add all residuals, we will find that this sum is smaller than the result obtained if we take any but the "average" value, as the correct one, and take the sum of the squares of the differences between any other value assigned to be correct and all the independent observations.

Or repeating (1) in another way:

(2) THE MOST PROBABLE VALUE OF ANY CONSTANT PHYSICAL QUANTITY AS DETERMINED BY ANY NUMBER OF EQUALLY CAREFUL INDEPENDENT OBSERVATIONS, IS THAT VALUE WHICH MAKES THE SUM OF THE SQUARES OF THE RESIDUALS A MINIMUM.

Just why we use the sum of the squares rather than the cubes or any other power will not be demonstrated here but it can be shown to be so.

For the present we are interested only in the practical appli-

cation of this theory to every-day engineering and naval life, and right here I might add, that if we made full use of our knowledge of this theory there would be big changes in range finder personnel in the fleet, spotters would lose their jobs—and—well we will see the application in a minute.

Let us go back to our problem of determining the base line. Perhaps you were not quite satisfied with the average result you obtained and sent your assistant out to remeasure the distance, and suppose in turn he brought in results closely comparable with yours but none agreeing exactly with your own. You might say—well, I'll average the whole lot and take the average of all as the true length. Just there you go wrong! Why?

Because as yet you have no absolute way of determining that both of these sets of observations were "equally careful" as is required by our first definition.

Even though your assistant's average length might coincide *exactly* with your average length, that is no indication that the observations were equally careful.

To exaggerate a trifle; suppose two people try to determine the absolute length of the standard meter, using a steel rule accurate to hundredths of an inch. One observer brings in measurements 39.40, 39.37, 39.34, averaging 39.37. The other brings in results 39.30, 39.40, 39.41, again averaging 39.37. Would there be any question in your mind as to which was the best observer? Certainly not! Look at the relative constancy of the two. The first has a maximum difference of .03 inch from his average; while the second has a maximum error two and a third times as large or .07 inch.

Naturally, I think you would give more credence to the results obtained by the first observer than to those obtained by the second, even though the average of both sets were the same. Just what the relative value of the two sets is might remain in doubt, but as will be shown a little later, the exact relative weights to be assigned is readily determinable.

Suppose we wish to use the result as obtained by our measurements in a problem involving multiplication or other mathematical manipulations and we must know about how accurate the result is going to be, and whether or not the accuracy is within the "limit of tolerance" required. In other words, we wish to know the "Limit of Errors" of the mean. If we know this, direct multiplication shows the limit of error of the solution. We might assume that since the maximum difference between extreme readings is 0.11 inch, the result could be no closer than this. If this is so, why did we take several readings?

We will find that the average or arithmetical mean is correct within a limit much less than the difference between extremes. In fact we will find:

(3) THE GREATER THE NUMBER OF EQUALLY CAREFUL OBSERVATIONS CONSIDERED, THE LESS WILL BE THE PROBABLE ERROR OF THE ARITHMETICAL MEAN.

What is the probable error of the average and how can it be obtained?

Without, at this time, going into the deduction of the theory, we may determine the probable error by an accurate formula by Bessel:

If we let r = the error of a single observation
and r_0 = the probable error of the mean
and n = the number of equally careful observations considered
and $/V_1/, /V_2/, /V_3/, \ldots /V_n/$ = the actual value of the residuals disregarding signs.

We may write, using the accurate Bessel Formula:

(4) Probable error of any single observation

$$r = \frac{0.6745}{\sqrt{n-1}} \sqrt{/V_1/^2 + /V_2/^2 \ldots + /V_n/^2}$$

(5) Probable error of mean

$$r_0 = \frac{0.6745}{\sqrt{n(n-1)}} \sqrt{/V_1/^2 + /V_2/^2 \ldots + /V_n/^2}$$

Note that in these formulae

$$(6)\quad r_0 = \frac{r}{\sqrt{n}}$$

These formulae look troublesome, but, fortunately, any good engineer's handbook tabulates the values

$$\frac{0.6745}{\sqrt{n(n-1)}}, \quad \frac{0.6745}{\sqrt{n-1}}$$

In practice, it is seldom that one has over ten observations, and if the work justifies it, sufficient accuracy is obtained by picking the value of r or r_0 from a curve. The accuracy of this curve, of course, depends upon the scale used.

Purely as a practical time saver you will find that if you accustom yourself to take either five or ten good observations, considerable time is saved. With five observations, take the sum, multiply by 2 and point off one place for the mean value. With ten observations, add all and point off one place. This, of course, applies equally well to angles. For example:

	Degrees	*Minutes*	*Seconds*
1st observation	62	52	10
2d observation	62	52	05
3d observation	62	52	12
4th observation	62	52	07
5th observation	62	52	14
	—	—	—
Neglect degrees and minutes			48
			2
			—
Multiply by 2 and point off			9.6

Average value 62 degrees — 52 minutes — 9.6 seconds.

If r as determined above is small, and by small I mean in comparison to the mean value, we have an indication that the observations are good, and accidental errors, or big errors of

observation have been eliminated, the systematic or instrumental errors may still remain. This phase is considered under the heading "What Errors Are."

To indicate the probable accuracy of a figure after determining the arithmetic mean and r_0 we write the result as follows

$$39.370 \pm .003.$$

WEIGHING THE MERITS OF OBSERVERS AND WEIGHTING THEM.

Few people realize the uses to which we could put this Theory of Least Squares in normal Naval life, and a very few will be pointed out.

In determining the length of the meter, described above, I intimated that one observer was better than the other, leaving how much better to be determined. It is a natural assumption to say that if we can accurately determine the relative accuracy of both sets of observations and then consider both sets in their relative values, the ultimate result is, in accuracy, greater than either set independently. That is, I must assign a correct weight to the arithmetic means.

Suppose it became necessary in a survey of the Virgin Islands to establish a long base line by astronomical methods. Naturally this would require extreme accuracy. One observer might have, and we know by long experience that most observers do have, a constant psychological error or personal error as it is called. Any determination of one end of the base by that observer would be affected by that error. With two observers, weighting their determinations independently, a better fix is obtained.

Again—aboard ship at the present moment the training of range finder operators is an important study. In time of war we must have the best operators available. Unfortunately, up to the present, the selection of operators has been more or less a case of favoritism based on personality and persistency, rather than on a systematic scientific analysis. No blame attaches to this method as the favoritism is not deliberate but is because no better method is generally understood. A man

is selected, let us say, and undoubtedly he is a good man. Training with him is another man whose personality is not quite so good and yet results may show that this is the better man psychologically, a man who will show steadier work and more consistent progress. How many of us can make a selection of men—determine their relative worth—mark their progress by scientific methods?

All this can be done by Least Squares! And there are hundreds of other applications of this theory to ordinary work, which will suggest themselves to the reader who has gone this far.

Let us then see how problems of this nature are worked out, and applied. Let us assume three practical problems covering

(*a*) The establishment of an astronomical fix.

(*b*) The selection of Range Finder Operators and determination of their relative worth, and improvement under training.

(*c*) The determination of constants in a formula whose type we know or suspect from data obtained on tests.

EXAMPLES IN LEAST SQUARES.

Problem I—Two observers, A and B, make a series of independent observations of the angle between two points from identical positions. The results are tabulated below. What is the most probable value of this angle? What is the probable error of this determination? Which is the best observer? What are their relative worths as observers?

Angles as determined by Observer A			Angles as determined by Observer B			
Deg.	Min.	Sec.	Deg.	Min.	Sec.	
62	42	50	62	42	50	consider as $60° - 42' - 60''$
62	42	40	62	43	00	
62	42	40	62	42	20	
62	42	50	62	42	30	
62	42	41	62	42	50	
Mean 62 deg.—	42 min. —	44.2 sec.	62 deg—	42 min —	42 sec.	

Residuals A		Residuals B	
$\lvert V\rvert$	$\lvert V\rvert^2$	$\lvert V\rvert$	$\lvert V\rvert^2$
5.8	33.64	8	64
4.2	17.64	18	324
4.2	17.64	22	484
5.8	33.64	12	144
3.2	10.24	8	64
23.2	112.80	68.0	1080.0

Probable error of mean $r_0 = .1508 \times \sqrt{112.8} = 1.601^s$ A—"Bessel"
Probable error of mean $r_0 = .1508 \times \sqrt{1080} = 4.956^s$ B—accurate

From this we see that while there is a probable error of but 1.601 in A's mean, there is a probable error of 4.956 in B's. We might assume that A is a better observer than B, but to assume that the relative efficiency was in the ratio of $\frac{4.956}{1.601}$ would be dead wrong.

It can also be shown that

(7) "THE WEIGHT TO BE GIVEN EACH SET OF OBSERVATIONS VARIES INVERSELY AS THE SQUARE OF THE PROBABLE ERRORS OF THE ARITHMETIC MEAN."

$$\text{or } \frac{\text{Weight to be given A's observation}}{\text{Weight to be given B's observation}} = \frac{(4.956)^2}{(1.601)^2} \text{ or } \frac{9.57}{1}$$

In other words, A's observations are almost ten times the value of B's.

The most probable value of the angle, therefore, taking into consideration both A's and B's readings, would be:

$$\frac{(9.57 \times 44.2) + (42 \times 1)}{9.57 + 1} = 43.99 \text{ seconds or } 44 \text{ seconds}$$

The most probable value of the angle is therefore

62 degrees — 42 minutes — 44 seconds.

How close is this value to the true value?

Neglecting degrees and minutes, and assuming 44^s as the best value we again take the residuals

	A			B		
		$/V/$	$/V/^2$		$/V/$	$/V/^2$
	50	6	36	50	6	36
$44^s\sim$	40	4	16	60	16	256
	40	4	16	20	22	484
	50	6	36	30	14	196
	41	3	9	50	9	81
$\Sigma/V/^2$			113			1053

Since A's observations have a weight 9.57 times B's, we must repeat each of A's observations 9.57 times or what is the same thing multiply $\Sigma/V/^2$ by 9.57.

$$9.57 \times 113 = 1081.41.$$

To this we add B's unweighted $\Sigma/V/^2$.

$$1081.41 + 1053 = 2134.41.$$

The number of observations manifestly has now changed and we are considering

$$9.57 \times 5 + 5 = 52.85$$

$$\text{or } n = 52.85.$$

Bessel's accurate value for $n = 52.85$ is .0129

and $r_0 = .0129\sqrt{2134.41} = .596$

or the angle is $62° - 42' - 44'' \pm 0.596''$

Problem II—Three range finder operators are put under training and after three weeks drills are examined for accuracy, using an object approximately 15,000 yards away. The results obtained are tabulated below. Three weeks later a second test is given and ranges obtained again tabulated as shown. Which is the best operator at the first test? What is the relative value of the operators? Which shows the most promise? What is the rate of improvement in each case?

RANGES OBTAINED BY OPERATORS.

First test.

	A	B	C
	15300	14800	15000
	14700	14900	14600
	15200	15000	15300
	15600	15100	15400
	15200	15200	15000
Mean	15,200	15,000	15,060

Second test.

	15300	14200	14900
	15400	14600	15000
	14600	15000	15200
	14700	15000	14900
	15000	15200	14800
Mean	15,000	14,800	14,960

Tabulating the residuals of each observer. we obtain

First test.

A		B		C	
$/V/$	$/V/^2$	$/V/$	$/V/^2$	$/V/$	$/V/^2$
100	10000	200	40000	60	3600
500	250000	100	10000	460	211600
000	000000	000	00000	240	57600
400	160000	100	10000	340	115600
000	000000	200	40000	60	3600
	420,000		100,000		392,000

Second test.

A_1		B_1		C_1	
300	90000	600	360000	60	3600
400	160000	200	40000	40	1600

600	360000	200	40000	240	57600
700	490000	200	40000	60	3600
000	000000	400	160000	160	25600
	1,100,000		640,000		92,000

You will note that averaging the ranges in the first and second tests shows absolutely nothing but the fact that in their present state of training the average is the "most probable range" for each individual.

Tabulating the residuals, however, is a very important step as now the relative worth of operators and their progress becomes apparent without further computation. By glancing at the squared residuals we see that on the first test the standings of the operators were

B, first; C, second; and A, third.

On the second test the standing is

C, first; B, second; and A, third.

By comparing the residuals of the first and second tests we see also a fact that is not apparent from a tabulation of the ranges taken, namely, that while C has shown considerable improvement, both A and B have gone backward.

In a letter from the Commander-in-Chief addressed to the Fleet occurs this sentence: *"determine the consistency and accuracy of each operator and do not hesitate to discontinue training of any man who does not give good promise. It is quicker and more satisfactory to train a new man."**

There would be no question apparently as to which operators should be retained or dropped. The rate of improvement, however, requires further analysis.

* In this same letter also occurs the following: "Keep records of performances of operators when ship is at anchor. Record mean of ten observations of same object, the difference between each observation and the mean, the average of these differences disregarding signs . . . ." In other words, the mean of residuals. This is a common error, as this mean of residuals has no mathematical meaning. There is in this letter one other statement which, considered from a standpoint of instrumental accuracy, is incorrect, namely, "to eliminate large variations in range finder errors . . . take ranges alternately from higher to lower and lower to higher ranges." As a matter of fact, the greatest instrumental accuracy is obtained when all readings are taken *in the same direction.*

The probable error of C's mean on first and second tests would be—

$$\text{Test 1 } r_0 = .1508 \sqrt{392000} = 94.42 \text{ yards.}$$

$$\text{Test 2 } r_0 = .1508 \sqrt{92000} = 45.74 \text{ yards.}$$

His improvement during the period of these two tests is mathematical expressed by the relation of $\left(\frac{r_0\ 1^2}{r_0\ 2^2}\right)$ or $\left(\frac{94.42}{45.74}\right)^2$ or 426 per cent improvement in accuracy.

A curve of improvement, from test to test, with time and per cent improvement, as coordinates, shows clearly and unmistakably the operators to keep and those to release. In such a graph there should be no sharp breaks but the slope should be negative and decreasing in value with time. It should be noted here that a computation of this kind does not show the actual range. What it does show is the most probable "range finder range," the instrumental error remaining. A calibration curve for each operator is still necessary.

From what has gone before, I believe the reader will realize that this theory has a good many practical applications. Many others will suggest themselves, as for example in determining mean point of impact from rake observations or from a photographic negative taken from a definite altitude; but probably the greatest use comes in surveying large areas where the curvature of the earth cannot be neglected and where spherical excesses require splitting among (angles obtained by observation) the three angles of a triangle or the four angles of a quadrilateral before it can be "closed."

To the Test Engineer, however, more than to any one else this theory is a necessity, especially if he be engaged in a research involving determination and evolution of laws.

Two specific cases only will be considered as all others are amplifications or variations of these two.

LEAST SQUARES AS APPLIED TO TEST ENGINEERING.

As a result of a series of tests, in which three quantities are capable of being varied at will, the Test Engineer after having plotted his experimental results on coordinate paper may suspect that the results of the test may be expressed by a formula in the shapes of

$$ax \pm by \pm cz = p$$

in which a, b and c are constants and x, y and z the unknown variables. The engineer would note in a long series of tests the value of p as he varied successively x, y and z. That is to say, he would have data for n equations. There being three unknowns, three equations only are necessary. Here, however, he has data for n equations all presumably equally accurate and no way of telling which are the best three to use. He also finds that none of his values when substituted give exact values of x, y and z.

We write, however, a series of n equations with our experimental data thus—

$$\begin{aligned}
a_1x + b_1y + c_1z &= p_1 \\
a_2x + b_2y + c_2z &= p_2 \\
a_3x + b_3y + c_3z &= p_3 \\
&\cdots\cdots\cdots\cdots \\
&\cdots\cdots\cdots\cdots \\
a_nx + b_ny + c_nz &= p_n
\end{aligned}$$

We now multiply each equation by the coefficient of x in that equation and add all equations to obtain what is called the normal equation in x.

Similarly we multiply each equation by the coefficient of y in that equation and add all equations to obtain the normal equation in y.

Similarly for z. We thus obtain three normal equations in x, y and z, which may be solved in the ordinary way. Note that any coefficient in any equation may be zero. The solution obtained as above is the best value obtainable.

Calling $aa = a_1^2 + a_2^2 + \ldots . a_n^2$

and $ab = a_1b_1 + a_2b_2 + \ldots . a_nb_n$

and $ap = a_1p_1 + a_2p_2 + \ldots . a_np_n$

We see that the three normal equations are—

$$(aa)\, x_o + (ab)\, y_o + (ac)\, z_o = (ap)$$
$$(ab)\, x_o + (bb)\, y_o + (bc)\, z_o = (bp)$$
$$(ac)\, x_o + (bc)\, y_o + (cc)\, z_o = (cp)$$

The numerical solution of these equations may prove tedious unless we use the trick of solving by determinants, writing a third order determinent for each unknown

$$\text{calling } D = \left\{ \begin{matrix} aa, & ab, & ac \\ ab, & bb, & bc \\ ac, & bc, & cc \end{matrix} \right\}$$

$$\text{then } x = \left\{ \begin{matrix} ap, & ab, & ac \\ bp, & bb, & bc \\ cp, & bc, & cc \end{matrix} \right\} \div D$$

$$\text{and } y = \left\{ \begin{matrix} aa, & ap, & ac \\ ab, & bp, & bc \\ ac, & cp, & cc \end{matrix} \right\} \div D$$

$$\text{and } z = \left\{ \begin{matrix} aa, & ab, & ap \\ ab, & bb, & bp \\ ac, & bc, & cp \end{matrix} \right\} \div D$$

If the plot indicates that there is but one unknown and the curve is of the parabolic form

$$ax - bx^2 - cx^3 \ldots . qx^n = p$$

we use a similar method. We vary x and note p.

We now multiply each equation by the coefficient of the x term and add, obtaining the first normal equation. We next multiply by the coefficient of x^2, adding to obtain the second normal equation and so on to the n^{th} normal. Ordinarily three terms are sufficient. The solution is then a question of simple algebra.

We have so far seen that the theory of Least Squares is really nothing but a study of errors of observation, and if the reader has borne with me thus far he may be interested in going further and seeing what errors are.

WHAT ERRORS ARE.

Errors of observation are due to two causes, instrumental or systematic errors, and true errors of observation, or "physiological errors."

Both of these type of errors may contain constants or variables.

For example, a measuring instrument may be "off" a constant amount, or it may be "off" amounts varying with the reading range. Absolute perfection is a physical impossibility and all we can do is attempt to reduce these imperfections to a minimum.

We may calibrate our instrument, whether it is a voltmeter, a tape, a range finder or a sextant. Such calibrations are usually accomplished by experts or trained observers to reduce the calibration errors to a minimum. Errors will always remain but for practical purposes we may, depending upon the care used in calibrating, consider the calibration as correct, and, all errors, using these instruments, then become errors of observation, with certain reservations. For example, in using a fine adjustment, whether it be the vernier of an optical instrument, the roller of a range finder, pair of micrometers, or a sextant, the final movement of the vernier just preceding "mark" should invariably be in the same direction, for any given set of observations. This eliminates "back lash" or other instrumental errors to a large extent, or rather tends to make them constant in amount. If a series of readings is to be taken of a varying quantity, as for example an electric potential, if possible an attempt should be made to take each set with the values increasing or decreasing throughout the set and not first with a rising voltage and then a falling one. This pro-

cedure tends to make the error of observation more nearly constant.

The errors of observation, and it is with these errors Least Squares deals primarily, are mostly physiological, depending on acuteness of vision, defects in vision, acuteness of perception, reflex actions and muscular lag or the period between the instant of "willing to do an act and its actual accomplishment." Defects of vision, reflex actions and muscular lag go ordinarily to produce the "constant error of observation." For example, a navigator may after long experience find that in his work he has a constant error of plus 1 minute 20 seconds in reading a sextant and correct his reading accordingly. He calls this his "personal error." Correcting for this error, however, does not entirely correct his observations. What he has done is correct his *personal constant error.* His personal variable error still remains. The variable physiological error is due to other causes. For example, the normal eye cannot ordinarily read or distinguish distance of less than 1/100th of a millimeter accurately. If a man's eyes are wrong, the perception is decreased and error increased. Now depending upon the scale of the instrument used the error which cannot be avoided even by the finest observer would, on this assumption, be in the vicinity of $\pm$ 1/100 times the scale of the instrument. There are, of course, other physiological causes for error, one of the greatest being nervous temperament. It is these last errors that the Theory of Least Squares corrects.

(1) Dealing then with these errors we might reason that the observer is *equally as liable to read too high as too low.* That is certainly a rational assumption.

(2) We may assume that *he is more likely to make a small error than a large one.* That is equally logical.

With the above assumptions, it can be proved, by a mathematical demonstration out of place in this article, that

(*a*) the most probable value of a quantity observed is that which will make the sum of the squares of the errors of observation the least,

(*b*) the most probable value of any quantity as determined by any number of equally careful observations independently taken is the arithmetical mean or average of such observations,

(*c*) the probable error of the arithmetical mean is equal to the probable error of a single observation divided by the square root of the number of observations.

THE RAY ROTARY FUEL OIL BURNER.
TEST ON A MARINE SCOTCH BOILER.

By R. C. Brierly.*

During the months of May and June, 1925, a test of three Ray Rotary Fuel Oil Burners was made, as authorized by the Bureau of Engineering at the request of the United States Shipping Board, on a single ended Scotch Marine Boiler at the Fuel Oil Testing Plant, Navy Yard, Philadelphia. Runs at various rates of combustion were made in order to determine the character of results obtainable with this type of burner as installed.

The burners installed for this test were manufactured by the W. S. Ray Manufacturing Co., San Francisco, Calif. The size burner used is known as the 250 horsepower size. Sectional view of burner is shown in Figure 1.

Practically all mechanical oil-burning devices for steam-power purposes are made up of two principal parts: the "atomizer," and the "register." The atomizer is the appliance used to break up the liquid fuel into minute particles; the register is a tuyere that serves the purpose of introducing the air supplied for combustion in such a manner that it mingles with the oil-fog. Nearly all oil burners used in marine practice are of the mechanical pressure type. The atomizers incorporate the principle of forcing oil at relatively high pressure through channels designed to impart a high rotary velocity to the oil and to break it up under the influence of centrifugal force and force of translation along the axis of the atomizer; the registers are cones or cylinders containing air admission areas.

* Test Engineer at fuel oil testing plant, Navy Yard, Philadelphia, Pa.

The burner under consideration is unique in its method of atomization and differs radically from the mechanical-pressure type of atomizer. The oil supplied to this atomizer drools upon the interior wall of a rotating cup; the whirling motion of the latter tends to throw drops off radially from the edge of the cup, which turns in a (as viewed from the furnace side of burner) clockwise direction. A strong blast of air introduced in a counter clockwise direction from the vanes of an air nozzle catches the oil which has been partially broken up by centrifugal force, mixes with it, and at the same time propels it into the furnace in the form of a cone of oil fog.

Description of Burner Parts: This burner, like other types of mechanical burners, is made up of an atomizer and a register, and also embodies in each unit a turbo-blower which not only rotates the atomizer cup and furnishes "atomization—assisting air," but also helps the register in its function of air admission and control. Description will, therefore, be made under three headings: (*a*) register, (*b*) turbo-blower, and (*c*) atomizer.

(*a*) *The Register:* Referring to Figure 1, it will be seen that the register itself is made up of two concentric flanged iron rings—the outer provided with bolt holes for securing it to boiler front plate, the inner sliding within the outer. As the faces of these rings are at an angle of 55 degrees from the flanges, the register is not truly cylindrical in form, but rather a much abbreviated hollow truncated cone. Both rings are provided with twelve square air admission areas. By adjustment of the inner ring, the size of these air admission areas may be regulated; the maximum opening per unit admission area being 3⅜ square inches. Passage of air to the furnace is from the inner face of the inner ring of register through the twelve air admission areas.

The center of the register is closed by a circular hinged iron door which is lined with plastic refractory. This door is supported by lugs at one side of the outer register ring, and is

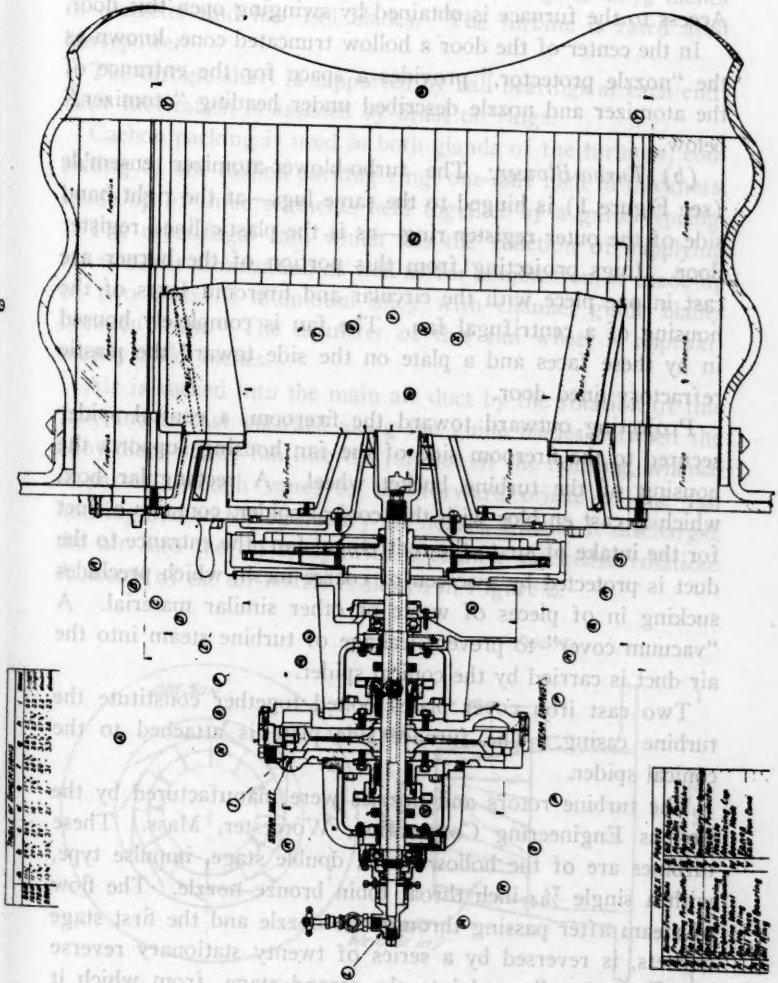


FIGURE 1.

held closed by a securing pin at the other side of register. Access to the furnace is obtained by swinging open this door.

In the center of the door a hollow truncated cone, known as the "nozzle protector," provides a space for the entrance of the atomizer and nozzle described under heading "atomizer," below.

(*b*) *Turbo-Blower:* The turbo-blower-atomizer ensemble (see Figure 1) is hinged to the same lugs—at the right hand side of the outer register ring—as is the plastic lined register door. Lugs projecting from this portion of the burner are cast in one piece with the circular and fireroom faces of the housing of a centrifugal fan. The fan is completely housed in by these faces and a plate on the side toward the plastic refractory lined door.

Projecting outward toward the fireroom, a conical spider secured to the fireroom side of the fan housing supports the housing of the turbine bucket wheel. A rectangular box, which is cast en bloc with this conical spider, contains a duct for the intake of air to the centrifugal fan; the entrance to the duct is protected by a screen of coarse mesh, which precludes sucking in of pieces of waste or other similar material. A "vacuum cover" to prevent leakage of turbine steam into the air duct is carried by the conical spider.

Two cast iron cover plates bolted together constitute the turbine casing. The furnace side plate is attached to the conical spider.

The turbine rotors and nozzles were manufactured by the Coppus Engineering Corporation, Worcester, Mass. These turbines are of the hollow shaft, double stage, impulse type, with a single 7/64-inch throat tobin bronze nozzle. The flow of steam after passing through the nozzle and the first stage buckets, is reversed by a series of twenty stationary reverse buckets and redirected into the second stage, from which it makes its exit into the exhaust chamber. The first stage of the turbine wheel is 13¼ inches in diameter to the edges of the

shroud, and has 142 blades; the second stage is 13⅝ inches in diameter and has 146 blades. The turbine is rated at 3 horsepower.

The turbine shaft is supported by ball bearings at each end; their lubrication is assisted by brass oil rings.

Carbon packing is used in both glands of the turbines, consisting of one carbon packing ring, one-half inch in thickness, made up of three segments held together by a garter spring.

The centrifugal fan, which has the function of supplying air to assist atomization as well as for combustion, is made up of two discs of aluminum alloy with channel guide blades between them. The diameter of this fan wheel is approximately 20¾ inches.

Air is sucked into the main air duct by the rotation of this fan, and is cast outward along the guide blades between the alloy discs of the fan into the interior of the housing, whence it passes through vanes to the forward orifice of the fan housing and to the stationary air nozzle, whence it discharges the air into the oil spray with a counter clockwise rotation. A sketch of the air nozzle is shown in Figure 2.

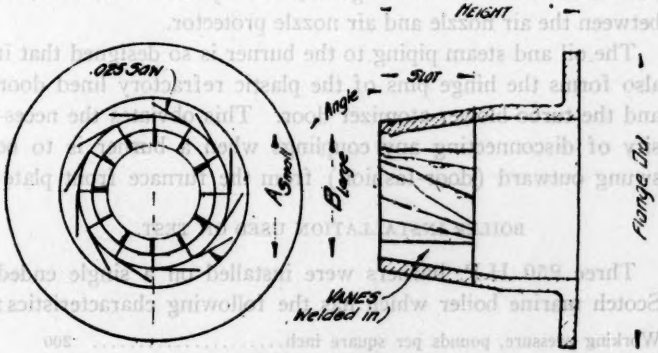


FIGURE 2—AIR NOZZLE.

SIZE 250 H. P., FLANGE 8 INCHES, SLOT 2½ INCHES, ANGLE 60 DEGREES HEIGHT 5½ INCHES, B. 4⅛ INCHES.

(*c*) *Atomizer:* The atomizer cup which rotates within the hollow center of the air nozzle is a brass cylinder, with the interior surface sloping outward at a very slight angle toward the delivery end which is slightly rounded. This atomizer cup is threaded on to the hollow shaft of the turbine and, like the centrifugal fan, revolves with it. Fuel oil reaches the interior of the atomizing cup via a ⅜-inch stationary tube, passing through the hollow shaft of the turbine. A ⅜-inch nozzle is threaded to the tube end, and from this the combustible drools upon the interior surface of the atomizer cup.

The functioning of the atomizer cup and the delivery of air from the vanes of the stationary nozzle has already been explained, but it is to be noted also that the strong blast of air making its exit from the air nozzle sets up a strong suction. This aspirator action induces a pronounced flow of air from the fireroom through the space between the furnace side of the fan housing and the fireroom face of the plastic refractory lined door, and thence into the furnace around the air nozzle exterior. Thus the air from the nozzle vanes is supplemented by additional air for combustion, not only through the air admission areas of the register, but by this induced flow between the air nozzle and air nozzle protector.

The oil and steam piping to the burner is so designed that it also forms the hinge pins of the plastic refractory lined door and the turbo-blower-atomizer door. This obviates the necessity of disconnecting any couplings when a burner is to be swung outward (door-fashion) from the furnace front plate.

BOILER INSTALLATION USED ON TEST.

Three 250 H.P. burners were installed on a single ended Scotch marine boiler which has the following characteristics:

Working pressure, pounds per square inch	200
Total boiler water heating surface, square feet	2555
Inside diameter, feet and inches	14–9
Distance between tube sheet, feet and inches	7–0 7/16
Overall length of boiler, feet and inches	11–0

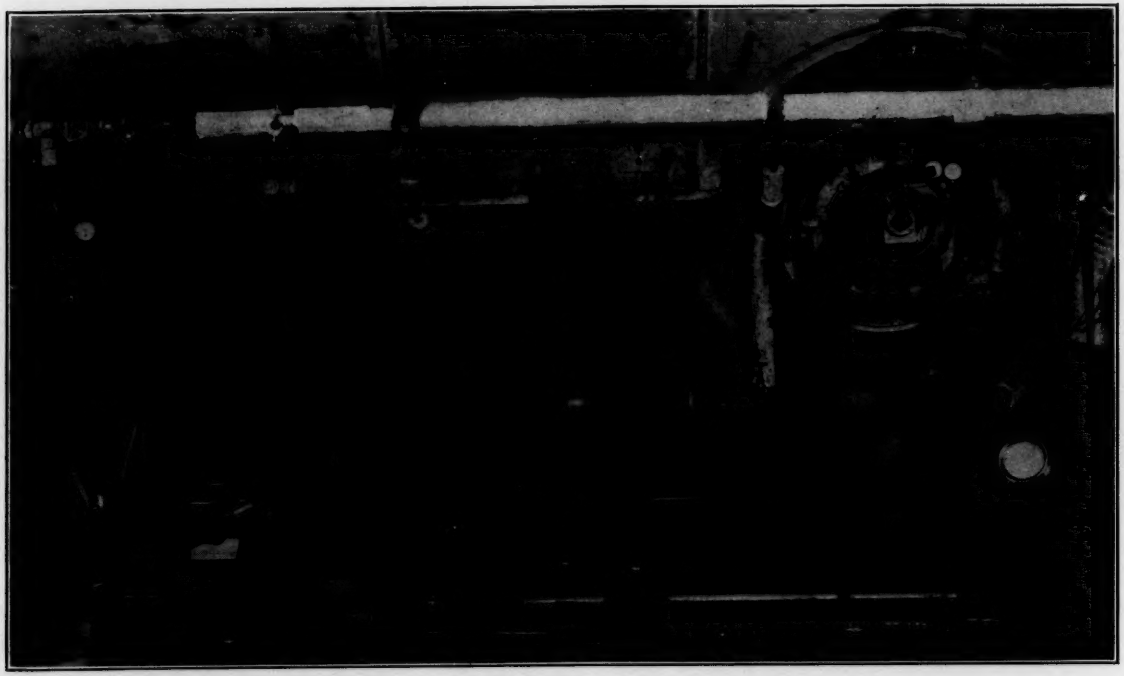


FIGURE 3.

Diameter of furnaces, feet and inches	3–7
Length of furnace, feet and inches	7–0
Total number of tubes	364
Volume of each furnace and its combustion chamber, cubic feet	161
Area of smoke pipe, square feet	7.41
Height of smoke pipe from center line of center furnace, feet	35.6

The boiler has individual combustion chambers for each of the three furnaces. The rear sheets of the combustion chambers are water cooled.

The steam pressure on the boiler and the main steam line from the boiler was maintained at a constant pressure by a hand controlled bleeder valve through which the steam exhausted directly to the atmosphere. A steam jet installed at the base of the stack was used to increase the draft when simulating various stack heights. A throttling calorimeter was used to ascertain the quality of steam. The feed water was weighed before being pumped to the boiler.

BURNER INSTALLATION.

The installation of three of the burners, one per furnace, is shown in Figure 3.

The entrance to each furnace was rammed up for a distance of 24 inches from the furnace front plate with plastic firebrick material, forming a refractory ring of 34 inches inside diameter. The outer rings of the registers were secured by studs to the furnace front plates. These rings carry a massive hinge flange with two lugs, which supports the plastic refractory lined door and turbo-blower assembly.

The upper pipe line which appears in Figure 3, just below the smoke box doors, is the steam manifold. At the left end of the line (as viewed in figure) were installed a strainer, a reducing valve for master control, and a by-pass with globe valves. From this manifold branch steam lines were led to each burner. After passing through the stages of the turbine, the steam made its exit through connections equipped with pop safety valves, set to lift at 9 pounds exhaust pressure.

Thence the steam passed through one passage of a double channel oil heating fitting to the "hinge-pin" system, and downward via the exhaust line into the exhaust manifold near the deck. Both ends of the steam manifold were furnished with steam gauges; steam pressure at entrance to each turbine was also determined by individual gauges. The exhaust manifold was equipped with a hand control valve for the maintenance of the exhaust pressure which was indicated by a compound gauge. This was done to simulate ship conditions where a positive pressure is maintained on the auxiliary exhaust line.

Figure 3 also shows three pipe lines below the burners; the lower pipe was the steam exhaust manifold; the upper and middle pipe lines were the oil manifold and the return oil lines. Oil could be by-passed through a return line to the service tank by opening the valve shown in the lower left hand corner of figure. Burner oil strainers were omitted during all runs.

The weight of oil used on the test was ascertained by oil weighing tanks, the oil passing thence to a constant level service tank. A motor driven Quimby pump was used as a fuel oil service pump. The oil pressure, at pump discharge, was controlled by a relief valve loaded with a beam scale. A hand operated valve at the entrance to the manifold ahead of the burners controlled the pressure beyond this point. The pump installation enabled oil to be furnished at the desired pressure, with a gauge fluctuation of less than one pound per square inch, when the combustion rate was constant. Oil passed from the service pump through oil heaters before going to the burners.

The oil pressure on the burner manifold was determined by a pressure gauge at the dead end of the line. The temperature of the oil to the burners was ascertained by a mercurial thermometer in a mercury filled well in the manifold before the connection to the first burner.

The path of oil from the burner manifold to the burners was through a ¾-inch brass pipe to the strainer containers

(strainers omitted); thence downward through the lower hinge pin fittings and upward to the tail pieces of the burners. For a distance of about four inches the oil entrance line paralleled the exhaust steam line, being slightly heated thereby. Before the tail piece valve, an oil pressure gauge was installed to determine oil pressure immediately at each burner.

TEST PROCEDURE.

The fire and water sides of the boiler were thoroughly cleaned before the test. As the burners were of a type with which the Fuel Oil Testing Plant staff were unfamiliar, they were lighted off for a preliminary run to ascertain the correct methods of operation. It was found that very low oil pressure delivered an ample amount of oil to the burners for low rates of combustion; but due to the difference in lengths of piping from the manifold to the burners, unequal amounts of oil were delivered to each. As the stop valves to the burners could not be adjusted finely enough to insure equal delivery to all burners, special plugs, $\frac{3}{8}$-inch long, drilled with $\frac{3}{32}$-inch orifices on the outlet side, and $\frac{1}{4}$-inch bore, $\frac{1}{4}$-inch deep on inlet side, were installed at the oil pipe coupling immediately before the tail piece valves. By this procedure, practically equal delivery to each burner was obtained.

Steam pressure to the burner turbines, during all the test runs, was regulated by a reducing valve. The oil used on all runs was unheated, it being taken at temperature of storage tanks, except during runs 7 to 11, inclusive, when it was precooled by passing water through the heaters, and on run 2 it was heated. During all the runs, unless otherwise noted, the following conditions were adhered to:

(*a*) maximum permissible smoke $\frac{1}{4}$ degree Ringleman;

(*b*) water level in right gauge glass 4 inches above lower nut;

(*c*) draft induced by steam jet regulated to produce a negative pressure of 0.3 inch at the rear of the furnace when

550 degrees F. stack temperature existed, the draft being increased as the temperature mounted, or decreased as it fell, by using a curve based on draft-stack temperature formula.

Before and after each run soot blowers were utilized to clean the fireside surfaces. When using soot blowers on a particular furnace, or when burners were not in use, the turbo-blower assembly was swung outward and the orifice in the plastic lined door was closed with a plug.

The method of adjusting the burners thought to be found most advantageous during the preliminary run—but later disproved—was to reduce the turbine speed until a trace of smoke from stack was apparent at the desired rate of combustion, with air-control registers wide open, then to raise the steam pressure to the turbines ten pounds and make adjustments of the air register proper to obtain a trace of smoke with CO_2 percentage approximately the same for each smoke box.

The exhaust pressure from the turbines was maintained constant at approximately 8 pounds per square inch gauge during all runs except numbers 13, 14, and 15.

Before run No. 7, the $\frac{3}{32}$-inch orifice plugs in oil line at burners was shifted to $\frac{1}{4}$-inch orifice plugs, as the very viscous oil fired during this and succeeding runs, necessitated a pressure at the burners of more than 50 pounds per square inch gauge to attain combustion rates above 400 pounds per burner per hour and it was desired to keep the oil pressure below 50 pounds.

The calorimeter was calibrated before the preliminary run, after run 6, and after run 15; on each occasion with rising and falling pressures, no steam leaving the boiler except through calorimeter—the main stop valve being secured. Thermometers were calibrated before the beginning of the runs; gauges, before beginning of runs, after run 6 and after run 15.

During each day's operation, runs were made advancing from a low to a higher rate of combustion.

FIG-4
RECORDED AND C
RUNS W
RAY ROTARY FUEL

RUN NO.	1	2	3	4	5	
DURATION OF RUN	1 HR.	1 HR.	1 HR.	2 HRS.	1 HR.	
NO. OF BURNERS	3	3	3	3	3	
AVG. NO. OF GAL.	3-647	3-647	3-762	3-768	3-768	
GRAVITY - DEGREE	18.9	18.9	18.0	18.4	18.4	
VISCOSITY - SAYBOLT UNIVERSAL	1188	1130	963	963	963	
B.T.U. PER POUND	18900	18900	19000	19000	19000	
FLASH POINT - °F.	190	190	240	240	240	
WATER - PER CENT	TRACE	TRACE	NIL	NIL	NIL	
CALORIMETER	396	396	396	396	396	
SATURATED STEAM AT BOIL PRESS.	388.7	388.8	388.4	388.6	388.9	
FEED WATER	208	209.4	209.8	212.5	212.6	
GASES LEAVING BOILER	534.5	542.5	552.5	537.0	564.0	
OIL AT BURNER	142.0	132.5	132.0	132.0	132.0	
TOTAL WATER EVAPORATED - POUNDS	145.58	159.53	164.97	328.83	331.07	
LBS. OF WATER EVAPORATED PER HR.	145.58	159.53	164.97	164.47	331.07	
QUALITY STEAM	99.69	99.69	99.69	99.69	99.69	
FACTOR OF EVAPORATION	1.0490	1.0510	1.0516	1.0460	1.0469	
EQUIV. EVAP PER LB. OIL FIRED	14.524	14.91	14.814	15.33	15.45	
EQUIV. EVAP PER LB. OIL AT 19000 B.T.U.	13.076	13.421	13.334	13.53	13.48	
TOTAL POUNDS OF OIL BURNED	1064	1164	1080	2584	1596	
POUNDS OF OIL PER HOUR	1064	1164	1080	1292	1596	
POUNDS OF OIL PER SQ. FT. H.S.	0.016	0.432	0.427	0.306	0.425	
POUNDS OF OIL PER CU. FT. F.V.	2.805	2.287	2.257	3.673	3.309	
POUNDS OF OIL PER BURNER PER HR.	354.7	388.0	360.3	431.0	532.0	
CO_2	12.5	11.96	11.76	13.06	13.40	
O_2	3.9	4.48	4.7	3.65	2.0	
CO	0	0	0	0	0	
N	83.6	83.6	83.5	83.9	84.2	
PER CENT EXCESS AIR	20.2	27.1	28.1	13.9	12.7	
BOILER STEAM	134	138.5	142.3	134.6	131.7	
STEAM TO TURBINES	76.6	74.9	72.9	95.0	136.0	
TURBINE EXHAUST	7.6	7.5	0.0	0.0	0.0	
OIL TO BURNERS	26.3	26.5	21.4	26.0	36.1	
PRESSURE AT BREECHING - INS. OF WATER	-0.26	-0.27	-0.27	-0.32	-0.38	
PRESSURE IN FURNACE - INS. OF WATER	-0.26	-0.25	-0.24	-0.27	-0.3	
BOILER - H.P.	444.7	457.0	471.0	584.4	774	
PER CENT OF RATING	172.0	176.2	182.7	227.6	279.4	
EQUIV. EVAP PER SQ. FT. H.S.	5.96	6.19	6.37	7.83	8.44	
UNAVOIDABLE HEAT LOSSES - %	11.30	11.33	12.36	12.68	12.47	
LOSS BY UNCONSUMED C - %	0	0	0	0	0	
LOSS DUE TO EXCESS AIR - %	1.56	1.82	1.86	0.78	0.69	
HEAT LOSS CHARGEABLE TO FURNACE & BURNERS -	11.60	11.80	11.16	6.14	8.70	
HEAT LOSS CHARGEABLE TO FURNACE & BURNERS	7.83	7.22	6.35	4.89	3.27	
HEAT LOSS CHARGEABLE TO H.S. - %	4.08	4.58	4.81	4.05	4.86	
OVERALL EFFICIENCY - H.S. FURNACE & BURNERS - %	76.77	76.72	76.87	78.51	78.83	
F.B. EFFICIENCY - %	74.1	74.8	74.0	76.8	76.8	
EFFICIENCY H.S. - %	85.84	84.53	84.05	85.37	84.06	
EFF. FURNACE AND BURNERS - %	91.16	91.74	90.74	90.11	90.76	
EFF. H.S. FURNACE & BURNERS COMBINED - %	86.68	86.54	87.24	90.49	90.45	

OIL

TEMP.-°F

FLUE GAS

PRESSURES

FIG-4
AND CALCULATED DATA
NS WITH
FUEL OIL BURNERS

5	6	7	8	9	10	11	12	13	14	15
1 HR.	1 HR.	8 MIN.	1 HR.	1-5 MIN.	1-5 MIN.	8 MIN.	0-5 HR.	0-5 HR.	0-5 HR.	0-5 HR.
3	3	3	3	3	3	3	3	3	1	1
3-748	3-748	3-779	3-778	3-778	3-773	3-778	3-813	3-793		
12.0	12.2	12.2	12.3	12.3	12.3	12.3	12.3	12.3		
963	968	9682	11863	6832	7813	6440	6800	6000		
8000	18000	18307	18307	18307	18307	18307	18307	18307		
840.	840	12.6	12.6	12.6	12.6	12.6	12.6	12.6		
MIL	MIL	0.2	0.2	0.2	0.2	0.2	0.2	0.2	[illegible]	[illegible]
306	306	306	306	306	306	306	306	306		
388.0	388.0	388.1	388.0	388.2	388.2	388.2	388.2	388.2		
212.6	210.9	211.5	211.3	211.4	211.8	211	210.7	211.6		
1044.0	440	441.2	451.4	438.6	381.3	1402	3783	3783	436.2	436.0
95.0	97.2	104.2	46.0	96.0	96.0	96.0	102.7	102.7	98.3	98.0
33707	18856	34.226		22106	32778	20008	4611	10107	2878	2838
33507	10038	10194		15467	21853	10402	13.770	14645		
99.88	99.76	99.86		99.9	99.80	99.82	99.84	99.82		
0.462	1.621	1.0498		1.0488	1.0455	1.0582	1.0926	1.0458		
0.43	14.48	14.82		14.82	14.72	14.84	15.0	15.71		
15.43	16.18	15.98		16.38	15.85	15.80	16.6	16.3		
1596	750	2542	781	1659	2254	1500	306	674	182	191
1596	750	1278.6	751	1098.7	1655	750	1448	1123	390.4	478
0.625	0.296	0.407	0.284	0.428	0.629	0.284	0.398	0.664		
1.305	1.556	2.618	1.556	2.268	3.282	1.584	2.41	2.35		
252.0	250.0	423.6	250.3	364.2	618.7	269	340.0	376.3		
13.00	12.7	12.1	13.2	13.7	13.6	13.0	13.2	13.2		
2.0	3.60	3.5	2.3	2.5	2.4	3.2	2.0	2.2		
0	0	0	0	0	0	0	0	0		
84.2	83.7	84.6	82.8	82.6	84.0	83.8	84.1	83.3		
12.7	11.0	12.0	13.8	12.4	12.8	12.8	13.3	13.5		
191.7	188.2	185.6	186.8	180.1	180.0	186.6	184.3	184.1		
360	80.0	144.0	78.0	155.0	159.0	78.0	72.50	146.0	152.3	110.6
0.0	0.0	0.0	0.2	0.0	0.0	0.0	0.0	0	0	0
36.1	34.0	33.1	33.6	32.1	31.4	33.0	40.7	41.0	32.7	40.2
0.38	-0.26	-0.34	-0.28	-0.29	-0.39	-0.28	-0.18	-0.06	-0.11	-0.10
05	-0.06	-0.31	-0.27	-0.26	-0.32	-0.20	-0.06	-0.08	-0.07	-0.06
714	389.0	853.0		468.4	694.6	346.1	487.2	611.2		
179.6	193.1	217		182.7	257.6	137.7	188.0	200.1		
2.46	4.45	7.47		6.54	8.97	4.87	6.3	6.9		
3.07	13.93	11.23		11.89	11.43	11.41	11.79	11.6		
0	0	0		0	0	0	0	0		
0.69	1.86	1.3		0.86	0.82	1.19	0.78	0.8		
3.70	6.35	9.45		8.98	10.58	11.37	4.47	5.15		
2.77	7.68	8.05		7.64	9.24	10.15	1.32	1.35		
4.88	1.49	3.48		2.44	4.88	1.32	3.45	3.8		
78.85	78.65	78.61		78.56	78.48	78.67	85.2	85.2		
74.8	75.8	78.5		79.3	74.0	78.6	78.0	78.8		
94.06	98.19	96.82		96.35	94.19	98.28	98.3	95.6		
96.76	96.77	98.81		96.40	98.63	96.84	98.6	98.8		
98.45	98.45	98.60		98.77	98.11	97.88	94.8	98.8		

RESULTS OF RUNS.

Results by individual runs may be seen by reference to Figure 4, which shows recorded and calculated data; by inspection of Figure 5, the comparative results of runs are shown graphically.

Referring to Figure 5, plotted from data of Figure 4, it will be noted that spots for runs 1, 2 and 3 do not fall in alignment with those of other runs; the main reason for their efficiencies being low is found in the magnitude of the losses chargeable to the Boiler Water Heating Surface. In this connection, it should not be overlooked that the average CO_2 percentages for these runs were relatively lower than those obtained on other runs of this series, the air registers and turbines not being regulated in a manner exactly similar, to that which was found best on later runs. It is believed that the lowness of CO_2 percentage and of the efficiencies in general on these runs was due to somewhat imperfect atomization, through inadequate turbine speed.

For purposes of general discussion, the data and spots of these runs,—1, 2, and 3—should be considered not as representative of possible efficiencies obtainable, but as indicating efficiencies perhaps obtainable by personnel only partially experienced in the operation of the burners. As conditions during each of runs 1, 2 and 3 were similar, except for changes in viscosity indicated in Figure 4, they may be considered comparable with each other, as far as operation at viscosities noted is concerned.

Efficiencies of boiler water heating surface alone, shown in Figure 5, demonstrated that the per cent efficiency of the B.W.H.S. was a straight line function of the oil burned per square foot of boiler water heating surface per hour, declining from approximately 98.2 per cent at the 0.294 pound rate (evidenced by spots of runs 6 and 11) to 94.4 per cent at the 0.625 pound rate (see spot of run 5). Spots of all runs except those for runs 1, 2, 3, 12 and 13, plotted well; numbers 1, 2

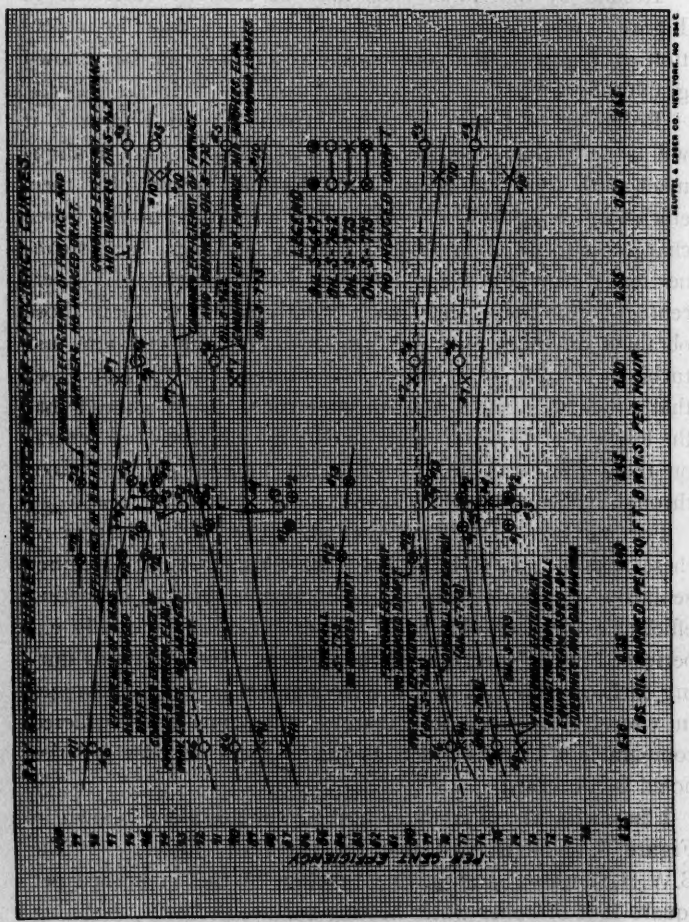


FIGURE 5.

and 3 failed to fall on the established line because of the reasons mentioned above; numbers 12 and 13 fell below the established boiler water heating surface efficiency line, because of the high heat losses with theoretical dry smoke pipe gases from saturation temperature at boiler pressure to temperature of gases leaving boiler. Conditions under which runs 12 and 13 were made are indicated in Figure 4.

Curves through the spots of combined efficiency of furnace and burners definitely established two outstanding results:

(1) that with these burners on this boiler, the percentage of combined efficiency of furnace and burners increased as the combustion rate was raised. As curves were still ascending at highest rate plotted, they showed that the combustion rate at which the burners themselves would attain their maximum efficiency was not reached on these runs;

(2) that the curve drawn through the spots of runs made with the less viscous oil at viscosity in the vicinity of 560 seconds Saybolt Universal was more than 2 per cent above the similar curve drawn through spots of runs made with the more viscous oils at viscosities in the region of 7000 seconds Saybolt Universal. The spots for runs 12 and 13, however, fell more than 4.5 per cent above the curve through spots plotted for runs made with the less viscous oil, although they were made with the more viscous oil. These combined efficiencies of furnace and burners of over 98 per cent were due to the extreme lowness of the summation of losses chargeable to furnace and burners, which were on runs 12 and 13 greatly diminished through reduction of losses due to unburned hydrocarbons.

Curves of the combined efficiencies of furnace and burners, eliminating unavoidable losses, shown in Figure 5, demonstrated that these efficiencies remained practically the same *for each oil* throughout the entire range of rates run.

Curve drawn for runs with the less viscous oil—at viscosity in vicinity of 560 seconds Saybolt Universal—was 2 per cent

FIGURE 5.

higher than similar curve drawn for runs with the more viscous oil—at viscosity in the region of 7000 seconds Saybolt Universal. These combined efficiencies of furnace and burners, eliminating unavoidable losses, of more than 94 per cent, were due to reduction of losses from unburned hydrocarbons.

The curves for overall (boiler and furnace) efficiency shown in Figure 5, demonstrated that while these efficiencies were slightly lower at the lowest rates of combustion at which runs were made than at the higher rates, they did not vary more than 1 per cent through the entire range of combustion rates run. The overall (boiler and furnace) efficiency curve with the less viscous oil lay above the similar curve for the more viscous oil, although the maximum advance in overall efficiency by use of the former was only approximately 1 per cent. The overall efficiency spots for runs 12 and 13 fell more than 4 per cent above overall efficiency curve obtained with the less viscous oil, although, as mentioned above, these runs were made with the more viscous oil, the high percentages of these runs were possible through diminished losses due to unburned hydrocarbons.

The curves for fireroom efficiency, calculated by deducting from the steam evaporated, the steam used by the turbines and the oil pump showed that the percentages of fireroom efficiency, while slightly less at the lower rates of combustion than at the high, did not vary materially throughout the range of combustion rates run.

Although overall efficiency curves obtained with the relatively less viscous oil versus the more viscous oil disclose that the viscosity of oil employed had little effect upon them, the more viscous oil required much greater speed of the turbines to atomize it, and, in consequence, a greater quantity of steam was used. More steam was also expended in pumping the more viscous than the less viscous oil. The fireroom efficiency curve for runs with the more viscous oil, therefore, falls considerably lower than that obtained with the less viscous. It

should not be overlooked, however, in this connection, that spots of fireroom efficiency of runs 12 and 13 fall higher than those for other runs with the more viscous oil—this because the better method of operation practiced during runs 12 and 13 raised the efficiencies so greatly that the high steam consumption necessary to run the turbines at the high speed employed, was more than counter-balanced by the great increase in equivalent evaporation obtained.

Had less viscous oil been employed in making runs which duplicated the conditions obtaining on runs 12 and 13, even higher fireroom efficiencies would probably have been possible.

GENERAL RESULTS.

The pressure required on the oil line from the pump was simply that necessary to secure the desired rate of oil delivery and could have been as low as 10 pounds gauge.

Oil of the highest viscosity employed on runs—11,060 seconds Saybolt Universal on run No. 8—was atomized easily.

Oil of viscosity in excess of 8000 seconds Saybolt Universal and containing 7.8 per cent water by volume in mechanical suspension failed—during run No. 15—to extinguish or interrupt the flame from the burner at 478 pounds per burner per hour combustion rate, or to cause an apparent change in atomization. The decrease in CO_2 noted was principally due to the dilution of flue gases by steam, and to the reduction in combustible delivered to the burner.

During runs 12 and 13 the unburned hydrocarbon losses were so reduced that, in spite of increased stack temperature and CO_2 similar to that on runs 7 and 9, all efficiencies were increased with the sole exception of that of the boiler water heating surface. This undoubtedly demonstrated that superior atomization was obtained with high turbine speeds, air control registers wide open, and furnace pressure only slightly below that of the fireroom. During these runs the rate of gas flow through the tubes was much greater than with jet-induced

draft and slower turbine speeds. The CO_2 percentage failed to show this; this apparent anomaly was due to increased volume of gas resulting from the combination of carbon and oxygen, greatly reducing loss from unconsumed hydrocarbons. The efficacy of operating after the manner employed on runs 12 and 13 is demonstrated by the fact that fireroom efficiencies obtained were more than 3 per cent above those obtained by operation with more draft than was required and lower turbine steam pressure.

By reduction of the exhaust steam pressure from 8 pounds per square inch gauge to atmospheric pressure, but maintaining the same steam chest pressure on the turbines, the oil burned per square foot of boiler water heating surface was raised from 0.399 to 0.440 pound or 10.25 per cent.

The burners were shown during exhibition by the manufacturer to be capable of good atomization from as low as 50 pounds per burner per hour to 525 pounds, or from rating down to 10 per cent.

The paralleling of the exhaust steam lines from the individual turbines with the oil entrance lines to each burner for the short distance at the hinge did little to diminish the oil viscosity below that present at the manifold before the burners.

DEDUCTIONS FROM TEST.

Atomization with this centrifugal air atomizer is not a function of the oil pressure, but of the turbine speed, and of the velocity of the air flow through the air nozzle.

Oil of viscosity in excess of 11,000 seconds Saybolt Universal is satisfactorily atomized. This figure represents the maximum viscosity used during test—not the greatest of which this burner is capable. The more viscous the oil, the greater the advantage of operating at high turbine speeds.

Burners are slightly more efficient when burning relatively non-viscous than with more viscous oil.

Oil containing 7.8 per cent water by volume in suspension can be burned with little difficulty.

The foremost advantages of this type of atomization are:

(1) The capability of satisfactorily atomizing oil of very high viscosity, high water content, and also oil sludge or oils containing residual or sedimentary material.

(2) Knowledge of the viscosity-temperature characteristics of the oil utilized for firing—indispensable for the most efficient results in operation with mechanical—pressure type atomizers—is unnecessary.

(3) Shifting of atomizer parts—tips and plugs—with changing rates, is unnecessary; great flexibility with varying load conditions is thus attained.

(4) High pressure pumps and piping are unnecessary.

The rate of firing per burner at which greatest burner efficiency would be obtained is greater than 525 pounds per burner per hour; hence the burners were oversize for the rates run.

The most efficient method of operation is to run the turbines at high speed with the air registers wide open, using stack damper for control of the draft at low rates; at maximum rates, the damper should be wide open. The minimum speed of the turbine for good atomization with relatively non-viscous oils (500 seconds Saybolt Universal at temperature of firing) is probably in the vicinity of 1800-2000 revolutions per minute; approximately 2300 revolutions per minute would be necessary for the more viscous oils (7000 seconds Saybolt Universal at temperature of firing). When atomizing at rates above 300 pounds per burner per hour maximum turbine speeds should be used. The general procedure for damper and turbine control above 300 pounds per burner per hour is, therefore, the damper should be opened to produce the required air flow through the register, in excess of that being delivered or induced by turbo-blower.

Because of the large number of variables entering into the selection of a size of burner for a given installation of this type, such choice should be made only under expert advice. The principal factors to be considered in the selection of the size of burners are: type of drive, power and speed of driving

unit, capacity of fan, diameter of air nozzle, angle of vanes in air nozzle, size of air nozzle protector, and area and distribution of air entrance ports in register.

The method of atomization and functioning of burner is such that it is particularly well adapted to constant speed motor drive.

THE SELECTION OF LUBRICATING OILS FOR WORKING CONDITIONS OF BEARING LOAD, SPEED AND TEMPERATURE.

By Lieut.-Comdr. G. B. Vroom, U. S. N.

In spite of the vast amount of laboratory research in the field of practical lubrication, there are very few data available for application by practical engineers, to their specific problems. The selection of a lubricant for a unit operating under conditions which vary within small limits offers little difficulty, since past experience with similar units will indicate the proper viscosity; and if this knowledge does not fall within the experience of the engineer or operator in charge of the unit, the manufacturers of oils are usually able to assist through the medium of recommendation charts, or the services of their field engineering staffs.

Recommendation charts are generalizations in the broadest sense, and cannot always take into account such variables as physical and mechanical conditions of the unit, the skill employed in operation or temperature ranges encountered in different localities and seasons. It is conceivable that plant engineers, marine engineers, and purchasing agents would prefer to purchase or specify oils in conjunction with the advice of a field staff maintained by many companies for that purpose.

All engineers afloat and ashore, interested in examining into the requirements of the units under their charge, are in need of a method by which they can analyze the mechanical factors of a given bearing, which affect the choice of viscosity (the most important characteristic of a lubricating oil), and mathematically arrive at an answer expressed in terms of *viscosity*. Such an answer gives something to work on, since one viscosity at a given temperature is definitive of the grade of oil most suited for the conditions analyzed.

It is essential, however, that the factors employed in the analysis be such that an extensive knowledge of the theory of lubrication shall not be required, and that the analysis shall not be academic or lean to laboratory exactness, but rather serve as a guide, and at the same time assure a good margin of safety. Obviously oil is cheaper than tie-ups and repairs and an increased friction loss is preferable to savings effected through the use of a light oil at the expense of a dangerously decreased margin of safety.

The viscosity of an oil, a property intimately connected with its ability to maintain a film between two moving surfaces, or one surface moving relatively to another, is affected by the *pressure* tending to squeeze out the oil; by the relative speed of the moving surfaces; and by the temperature of the oil, the viscosity decreasing as the temperature increases. (See Figures 3*a*, 3*b* and 3*c*.)

Therefore, if we can express mathematically the relation of pressure and speed to viscosity value, and know or assume the temperature at which that viscosity value should be considered for a given bearing, we can estimate within practical limits what the viscosity characteristics of the oil best suited to that should be.

There is such a formula, deduced by Professor Moore, quoted and used by Gill in his chapter on "Lubrication" in Day's Handbook.[1] The formula is

$$P = K \sqrt{v} \quad . \quad . \quad . \quad . \quad (1)$$

P in (1) is the load, or dead weight on the bearing expressed in *pounds per square inch of projected area multiplied by a factor of safety.*

(*a*) In the case of turbine and similar bearings, the load on the bearing is the portion of the weight of the rotor and shaft supported by the bearing. Thus, in a turbine whose rotor and shaft has a total weight of 10,000 pounds, supported by two bearings, the load per bearing is 5000 pounds. If the bearing

[1] Handbook of the Petroleum Industry, Day: Vol. II, pp. 635, 648 and 649.

is 10 inches long, diameter 5 inches, the projected area is 50 square inches and the *load* per square projected area is 100 pounds.

(*b*) In the case of crank pin bearings of reciprocating engines, the load is the sum of the dead weight of the reciprocating parts supported by the bearing, and the *average* pressure on the bearing due to the thrust exerted by the piston. (The *average* pressure is taken since the *maximum* is exerted for but a fraction of a second.)

(*c*) The *factor of safety,* which is discussed later, is used with reference to the resistance to rupture of the oil film, and is *not* the factor of safety used in designing the bearing.

V is *surface speed* of the journal in *feet per minute.* It is equal to $\frac{\pi D}{12} \times$ R.P.M., where D is the diameter of the journal in inches, and R.P.M. is its *rated* speed.

K is a factor covering fit of bearing, clearance, smoothness, oil grooving, and the *influence of viscosity.*

Assuming that the bearing has been well designed and run in, the dominating part of K is the *influence of viscosity.*

Load and rated speed are constants embodied in the design of the machine. For a given bearing, therefore, we can find K from equation (1).

$$K = \frac{P}{\sqrt{v}} \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Having found K from equation (2), it remains to find the *influence of viscosity.* The graph reproduced here as Figure 1 shows the relation of K to *viscosity* expressed in *seconds Saybolt Universal.*

For the calculation of P, Gill gives a table[2] showing factors of safety in which the argument is *speed of journal, feet per minute.* This "factor of safety" table is not reproduced here, for the reason that the values are considered to be too low.

[2] See foot-note 1 *ante.*

In the application of equation (2), therefore, the steps are:

(*a*) For the given bearing, find P.

(*b*) $V = \frac{\pi D}{12} \times$ R.P.M. (R.P.M. is the rated speed).

(*c*) Find K from (2). $\left(K = \frac{P}{V v}\right)$

(*d*) From Figure 1 find viscosity for the value of K.

The viscosity thus found is assumed by Gill to be the viscosity necessary *at the operating, or assumed, temperature conditions* of the bearing, and this temperature, and the viscosity found as above, are taken by him as determining the viscosity-temperature characteristics of the oil required.

The writer has applied the methods above and the "factor of safety" table, as described by Gill, to a number of bearings, and in each case the viscosity-temperature determination deduced was far below that of present practice in the lubrication of internal combustion engines, high speed turbine bearings and similar machinery. Applying Gill's methods and factor of safety to a heavy high speed bearing, the result shows an oil, if the assumed temperature is the *average* running temperature of similar bearings, the viscosity of which is below that of a refrigeration or spindle oil—obviously not suited, in practice at least, to such a bearing. It does not allow, for example, for any rise in temperature above normal due to conduction; to variation in temperature of surrounding air; or for any abnormality, such as failure of circulating water supply, constriction of oil supply to bearings, or other casualty. Furthermore, it specifies an oil theoretically suited to *normal* running conditions only, and does not take into account variation in speed, or starting and stopping. Therefore, in order to bring results from the use of this method into line with accepted practice, which is known to give satisfactory results, and to allow for abnormalities in temperature rise and other variable operating conditions, two changes in the methods described should be made.

(*a*) In calculating P, a factor of safety should be used as follows:

Surface Speed, feet per minute	*Factor of Safety*
1000	10
1500	12
2000 and up	15

(*b*) The temperature selected to correspond to the viscosity found from K should not be an actual or assumed running temperature, but an upper limiting temperature; 180 degrees F. Most bearings run at lower temperatures than 180 degrees, and the viscosity will, of course, be greater than absolutely necessary. The friction loss due to use of a heavier oil is negligible in comparison with the dangers inherent in using an oil that would fail under abnormal conditions, existing even momentarily.

The application of Equation (2) is illustrated as follows, using the factors of safety shown above, the relation of K to viscosity, and an upper limiting temperature of 180 degrees F.:

Load on bearing (pounds per square inch projected area) = 66.6.

Surface speed, feet per minute = 6480.

From table of factors of safety, select 15.

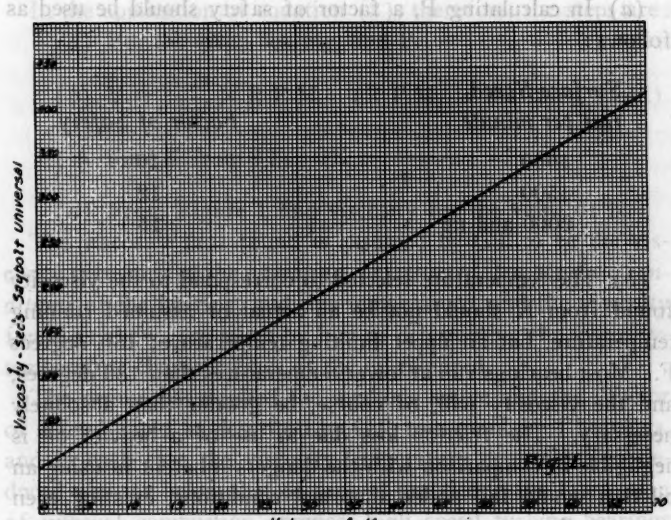
$P = 66.6 \times 15 = 1000.$

$v = 6480.$

From (2). $K = \frac{P}{\sqrt{v}} = \frac{1000}{6480} = 12.5.$

Enter Figure 1, and find, for K = 12.5, viscosity = 75 seconds. This viscosity is to be taken at 180 degrees F. the upper limiting temperature.

An oil having a viscosity of 75 seconds at 180 degrees F. would have from typical viscosity curves, a viscosity of 340 seconds at 100 degrees F.



From Day's Handbook of the Petroleum Industry Vol. II

II

DETERMINATION OF K AND VISCOSITY BY DIAGRAM.

To facilitate the solution of the equation $K = \frac{P}{\sqrt{v}}$, an alignment diagram has been constructed, Figure 2. Select the value of v in the right hand column; place a triangle or rule so that the edge passes through this point (value of v) and the origin at the lower left corner of the diagram. Select the value of P on its scale. The distance on the P scale, from the point where it is cut by the straight edge, to the value of P, represents log K. (Log K = log P − ½ log V.) K may be read off directly either (1) by pasting a logarithmic scale on the straight edge and reading off the value of K, or (2) by picking off this distance (log K = log P − log ½ v) with a

pair of dividers, swinging this distance at right angles to the P — scale from P. K is shown along the top of Figure 2.*

Further, the viscosity values corresponding to values of K shown in Figure 1 have been added as an additional scale to

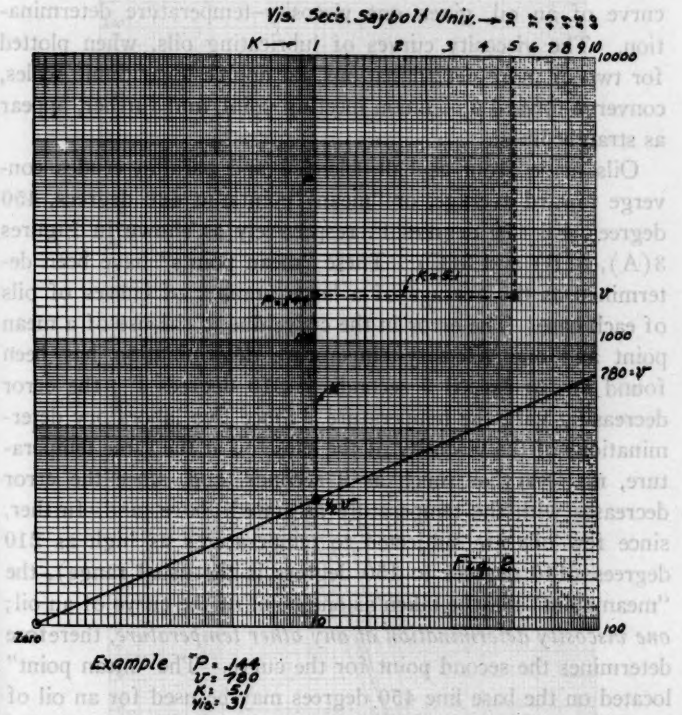


Figure 2; *therefore, given P and v, Figure 2 may be used to find the required viscosity reading, by one operation.* The arithmetical solution of the equation $K = \frac{P}{\sqrt{v}}$, and the use of Figure 1 is avoided.

* Examination of the example shown on the alignment chart (Figure 2) shows that heavy bearings, at low speeds, require a higher viscosity oil than the same bearings at higher speeds; also that light high speed bearings require relatively low viscosity oils—points that have been borne out by observation and in practice.

III

DESCRIPTION OF THE VISCOSITY-TEMPERATURE DIAGRAM.

Figure 4 is a viscosity–temperature diagram similar to that described for use with Fuel Oil[8] to determine the viscosity curve of an oil, given one viscosity–temperature determination. The viscosity curves of lubricating oils, when plotted for two or more determinations, on double logarithmic scales, converge toward a common meeting point, and further, appear as straight lines.

Oils made from asphalt, blended and paraffin crudes converge toward average, or "mean" points, at 420 degrees, 450 degrees and 470 degrees F. respectively as shown in Figures 3(A), 3(B) and 3(C). These "mean points" have been determined as the average of a large number of curves of oils of each base. The error in the curve due to the use of a mean point and one viscosity-temperature determination has been found not to exceed 9 seconds at 210 degrees F.; the error decreasing with the temperature. Since two laboratory determinations of the viscosity of the same oil at the same temperature, may vary as much as 9 seconds, and, since the error decreases with the temperature as stated above, and, further, since few oils are subjected to temperatures as high as 210 degrees (110 degrees to 150 degrees is the usual range), the "mean point" may be taken as one point of the curve of an oil; *one viscosity determination at any other temperature,* therefore determines the second point for the curve. The "mean point" located on the base line 450 degrees may be used for an oil of any base, without appreciable error, and is sufficiently close for practical lubrication.

IV

Referring now to Section II, having obtained the viscosity value corresponding to K, we assume this viscosity to be correct for the upper limiting temperature 180 degrees F. The "mean

[8] Journal American Society Naval Engineers, Nov., 1925, "Fuel Oil Viscosity."

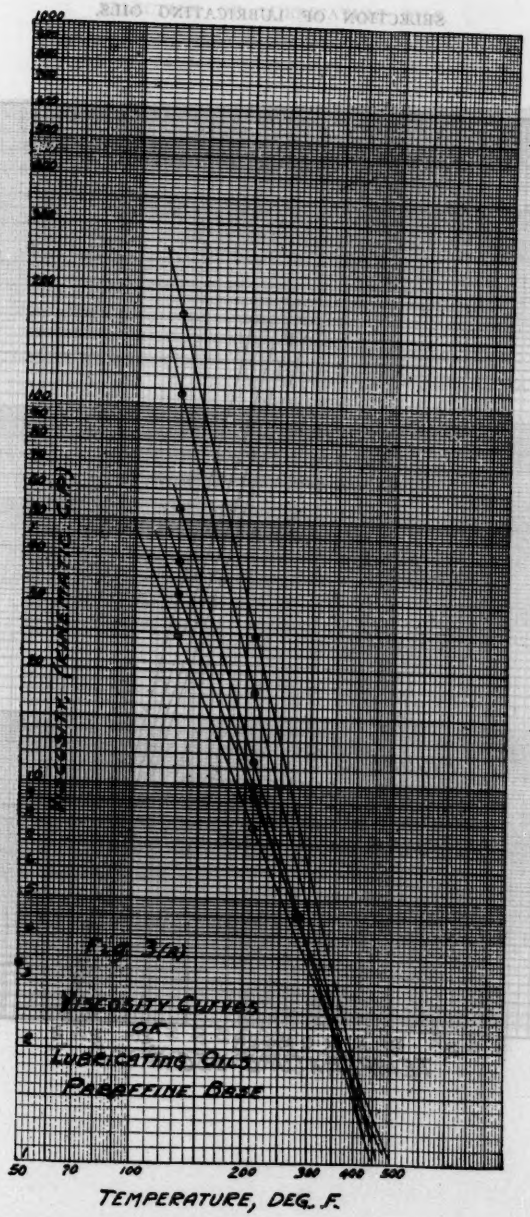


Fig. 3(a)

VISCOSITY CURVES OF LUBRICATING OILS PARAFFINE BASE

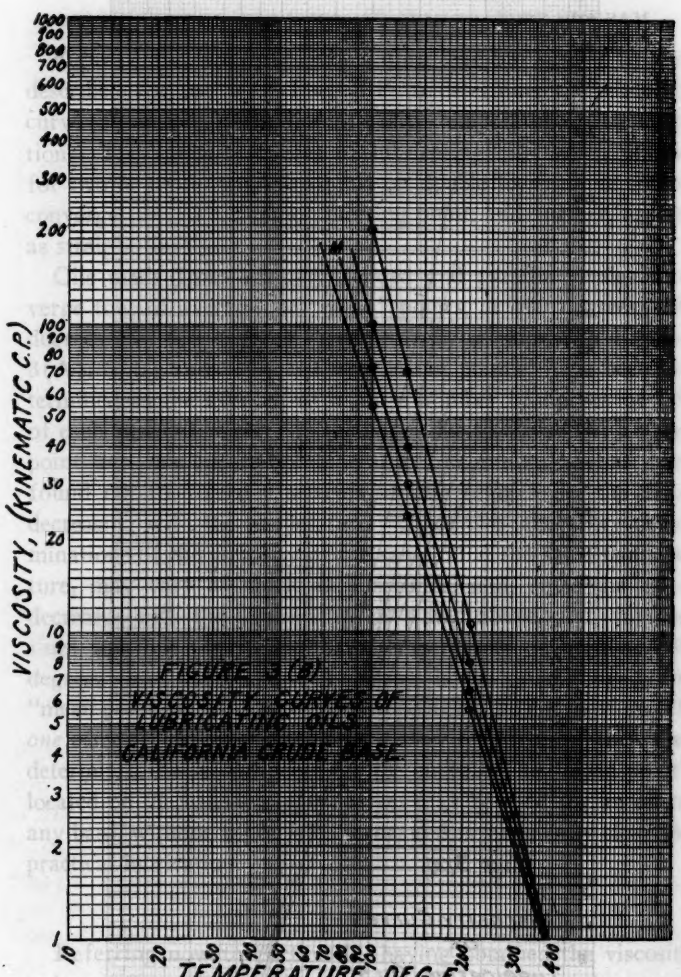
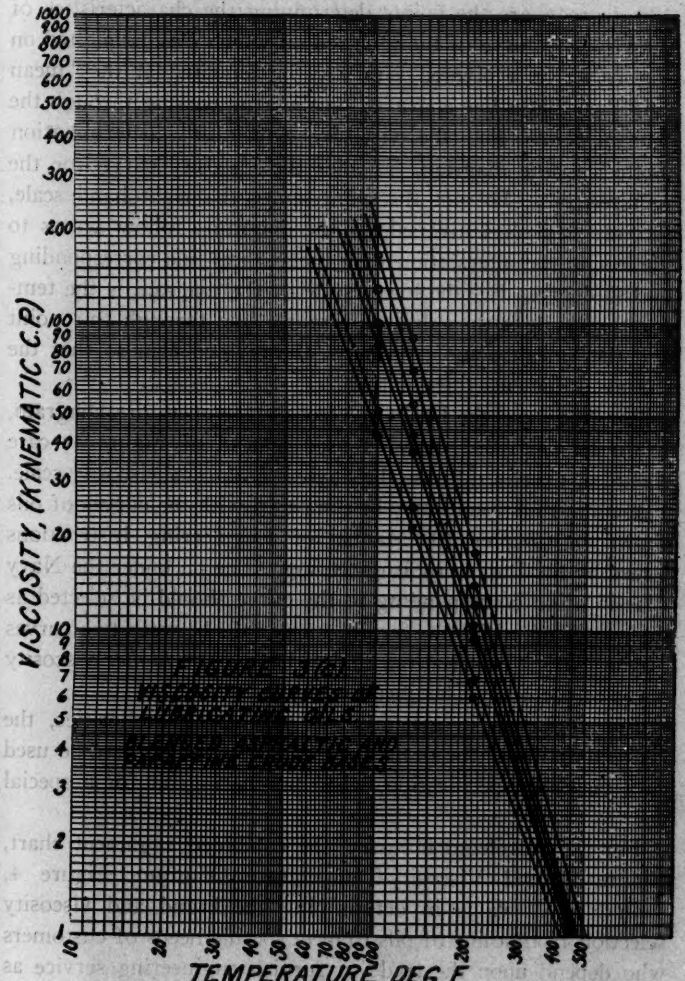


FIGURE 3 (B)
VISCOSITY CURVES OF LUBRICATING OILS.
CALIFORNIA CRUDE BASE.


VISCOSITY, (KINEMATIC C.P.)
1000
900
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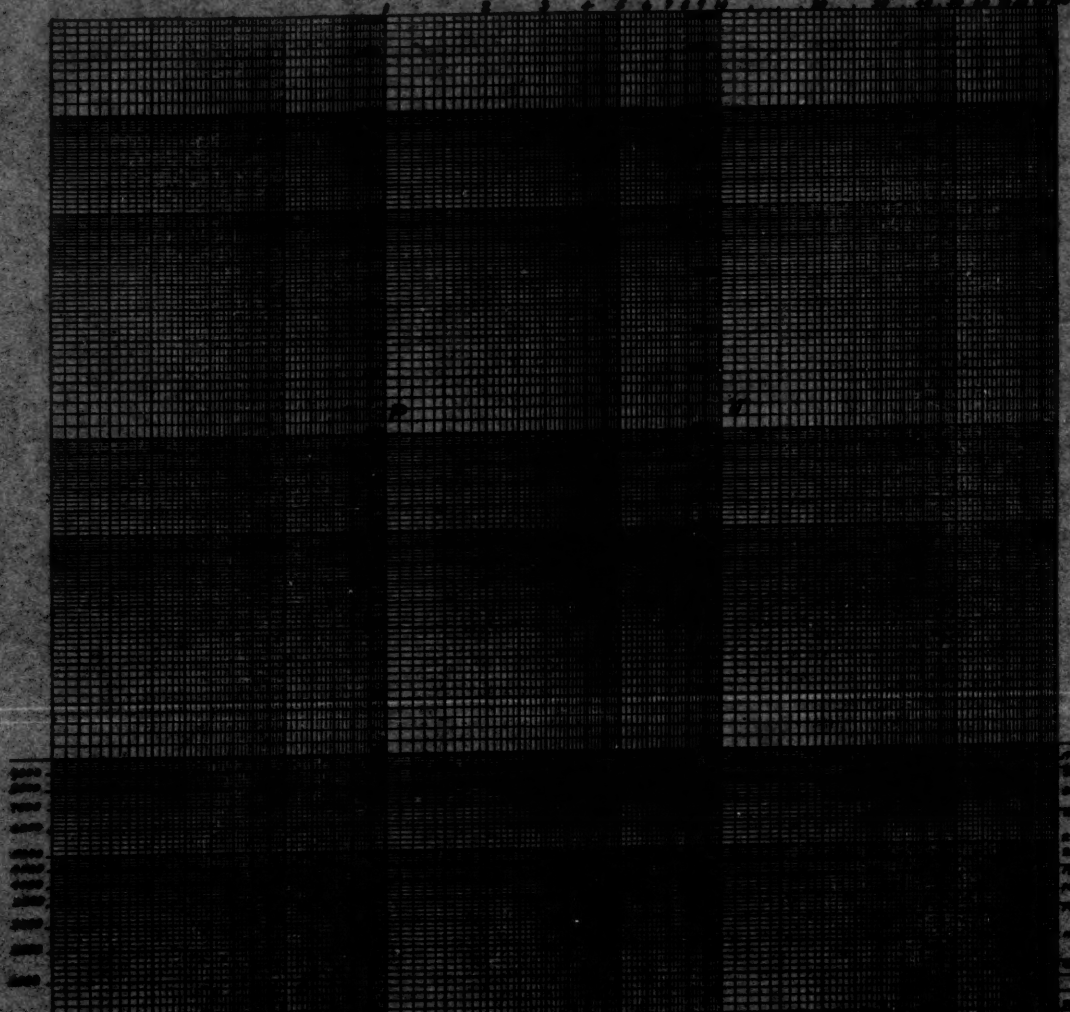
500
TEMPERATURE, DEG F

point" of Figure 4, and the viscosity (corresponding to K) at 180 degrees, are the points determining the characteristics of the viscosity curve of the required oil which may be plotted on Figure 4; or a pivoted arm may be mounted at the "mean point" so that a line engraved on the arm, passing through the "mean point" and the viscosity–temperature determination plotted on the diagram, becomes the viscosity curve for the required oil. To plot the point, enter the right hand scale, seconds Saybolt Universal, of the diagram; follow across to the left to the intersection with the vertical line corresponding to 180 degrees F. (the lower scale of the diagram is the temperature scale); then draw a straight line through this point and the "mean point"; or move the pivoted arm so that the engraved line cuts the point.

For Navy use, the Fuel Oil Viscosity-Temperature Diagram, Form N. Eng. 264, may be used instead of Figure 4; care should be exercised to use the Saybolt *Universal* viscosity scale. The curve thus found may be compared with the curves of oils shown in Chapter 10, Manual of Engineering Instructions issued by the Bureau of Engineering. The oil under the Navy Classification corresponding to the curve should be selected as the proper oil. If, however, the curve falls between two curves of the Navy Classification, the oil having the ***higher*** viscosity curve of the Navy Classification should be selected.

For engineers in general, and purchasing authorities, the characteristics of the desired oil thus determined may be used as a viscosity specification, when purchasing oil for a special use.

For the petroleum industry, the combined alignment chart, Figure 2, and Viscosity-Temperature Diagram, Figure 4, should be of value in determining the brand and viscosity selection from a line of oils to meet specific needs of customers who depend upon the trade to furnish engineering service as well as material.



$K = \frac{P}{V^2}$

Where P = load per sq. in. of projected area x K (factor of safety) - load up to 1200 lbs/in. sheet Ref 15

V = Running speed feet per minute.

MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS
STANDARD REFERENCE MATERIAL 1010a
(ANSI and ISO TEST CHART No. 2)
DENSITY Kinematic (cm²)
DENSITY Saybolt (universal)

V3812
MAY 2
XU

Viscosity Nomenclature
Engler
Chart
for
Lubricating Oils
Viscosity Saybolt (Universal)

MAY 21 1934

THE AIRCRAFT BUBBLE SEXTANT, TYPE A.

By COMMANDER R. C. PARKER, U. S. NAVY.

The Aircraft Bubble Sextant, Type A, was constructed from plans submitted by the Bureau of Aeronautics to fill the requirements of an instrument with which altitude of heavenly bodies could be measured without depending on a sea horizon.

While intended primarily for the use of aircraft, it will undoubtedly be found useful to surface vessels as well, since it eliminates the long period between evening and morning twilight when there may be stars in plenty but no horizon, and those other occasions so familiar to every navigator when the sun could be snatched through rifts in the fog if only there were some horizon to bring it down to.

Briefly, its principle consists in the substitution for a sea horizon of a self-contained artificial horizon, in the shape of a bubble with which the heavenly body is brought into coincidence. This principle is not to be confused with that of a theodolite, in which the instrument itself has to be adjusted to a true and steady level, nor with that of the ordinary mercury horizon in which the image of the heavenly body is made to coincide with its reflection in the liquid. In the bubble sextant the sun or star is brought down into coincidence with the bubble itself, and, within certain limits, is not dependent on holding the sextant exactly level.

The following are some of the outstanding features of its construction:

There is no graduated arc and no index bar with vernier, as in the ordinary type. The index mirror is moved by an arrangement of worm and gearing from a knurled thumb wheel, and the angle measured by its movement is recorded by a counter gear much like that on an ordinary speedometer or cyclometer.

In place of the horizon glass there is a glass bubble-cell like a round spirit-level, and the image of the bubble, by an arrangement of prisms and lenses, is projected through the index mirror and to the eye as though it were coming from an infinite distance.

In order to use the instrument in the ordinary manner on the sea horizon, as well as to check its accuracy, there is an object-glass or lens mounted in a horizontal tube beyond the bubble, which projects the image of the sea horizon or distant object through the bubble and index mirror to the eye.

The arrangement of optical parts is shown as a side view in Figure 1. B. is the bubble cell, filled with Zylol except for a small bubble at the top. The top and bottom of the cell are of glass and hence the whole cell is transparent, except that the bubble itself is projected to the index mirror, M, as a dark ring with transparent center. Light is thrown through the bubble cell either from the outside through lens (L_1), and right angle prism (R), or from the small internal electric bulb (X_1). The image of the bubble, reflected in the prism, and collimated in the lens (L_2), into parallel rays as though from a distant object, is projected through the transparent index mirror (M), and to the eye (E).

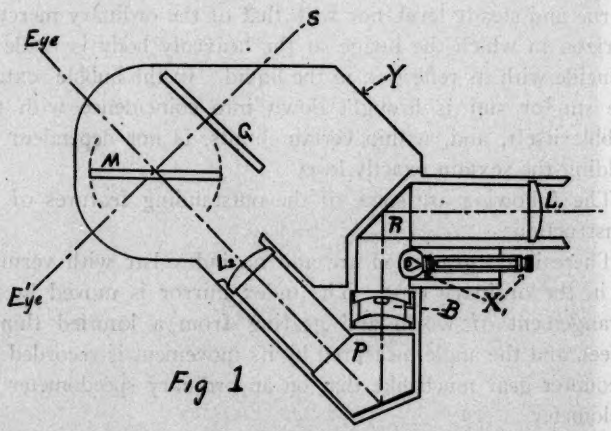


Fig 1

The index mirror (M) is turned until the reflected image from the sun or star (S) coincides with the projected image of the bubble, and the resulting angle is read off from the counter.

The two lenses, (L_1) and (L_2), together form a 1-power telescope through which the sea horizon or distant object is seen and may be used for coincidence with the sun or star in the ordinary manner; and also for adjusting the instrument, as will be explained later. The lenses are adjustable so that all parallax may be removed. A cap may be placed over the end of the telescope when it is desired to exclude terrestrial objects from the field.

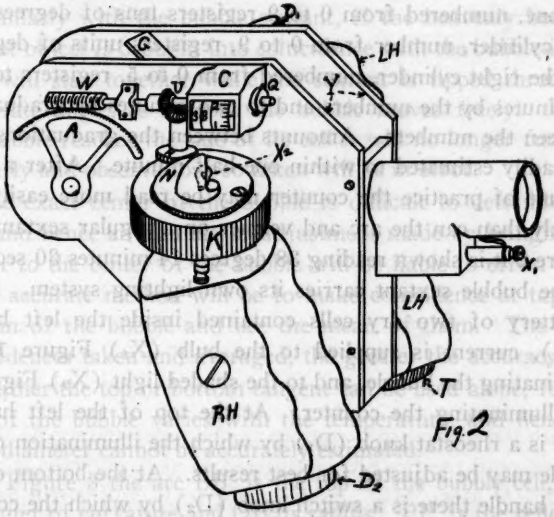


Fig. 2

Colored sun glasses, (G), may be turned into position between the index mirror and the sun.

All of these optical parts are carried between two parallel frame plates (Y), about an inch apart; the lenses, prisms and bubble cell being inside a tube, between the plates.

Outside of the frame plates (see Figure 2) is a handle or grip on each side, (RH) and (LH). At the top of the right handle is a knurled thumb wheel (K), which may be turned by the thumb and first fingers of the right hand to make the coincidence, while maintaining a grip on the handle, (RH). This thumb wheel, through two gear wheels, (N, N), a pair of bevel wheels, U, and a worm and arc (W, A), turns the index mirror (M), and also the shaft of the counter (C).

The counter and the worm are on the same shaft, so any lost motion in the gear and bevel wheels will not affect the readings of the counter and any between the worm and arc and in the counter itself is reduced to less than 1 minute.

The counter (C) consists of three graduated cylinders. The left one, numbered from 0 to 9 registers tens of degrees; the next cylinder, number from 0 to 9, registers units of degrees; and the right cylinder, numbered from 0 to 5, registers to tens of minutes by the numbers and to two minutes by graduations between the numbers. Amounts between the graduations may be readily estimated to within one-half minute. After a small amount of practice the counter may be read more easily and quickly than can the arc and vernier on a regular sextant. In Figure 2 it is shown reading 38 degrees 14 minutes 30 seconds.

The bubble sextant carries its own lighting system. From a battery of two dry cells contained inside the left handle (LH), current is supplied to the bulb (X_1) Figure 1, for illuminating the bubble, and to the shaded light (X_2) Figure 2, for illuminating the counter. At the top of the left handle, there is a rheostat knob (D_1) by which the illumination of the bubble may be adjusted for best results. At the bottom of the right handle there is a switch knob (D_2) by which the counter lamp (X_2) may be turned on or off. A cap, (T), at the bottom of the left handle, permits the removal of the dry cells.

Other features of the instrument are a watch holder on the back, to facilitate the observer in marking the time of his own sights, and a knob and pointer on the left side plate (not shown) by which the sun glasses may be set normal to the approximate altitude of the sun.

The instrument weighs about 2½ pounds, and is somewhat smaller than a regular sextant.

The design of the instrument requires that the observer hold it low, looking down into it at an angle of about 45 degrees, except when observing a star, which will be explained later. The field of the telescope with bubble in it, will be seen projected up through the transparent index mirror, the bubble as a dark ring moving about the field as the instrument wavers.

In an ordinary sextant, after the sun or star is in coincidence with the horizon, you can tip the instrument towards or away from them without causing them to separate. They will cross the field together, and an altitude can be taken in the top or bottom of the field as accurately as in the center.

Similarly with the bubble sextant, if the heavenly body or distant object is brought into coincidence with the bubble-image they will stay together while the sextant is tipped through a moderate angle, as long as the bubble moves freely. When the bubble reaches the side of its cell it will no longer function properly and observations so taken will be in error.

The exact center of the bubble is difficult to determine by eye, and hence an altitude or adjustment made by bringing an object to the center of the bubble will be liable to error. The more accurate method will be to make coincidence at top and bottom of the bubble and use the mean of them. The more coincidences taken and averaged, the greater the accuracy.

Neither the top or bottom tangent can be used alone, for the size of the bubble varies with the temperature, and hence its semi-diameter cannot be accurately estimated.

In Figure 3 the arc BB′ is the top of the bubble cell, O is its center of curvature and OB its radius. PP′ is the reflecting surface of the penta-prism, and F the point conjugate to O with reference to PP′, that is, the point such that lines drawn from any point in PP′ to both O and F will be equal and make the same angle with PP′. MM′ is the top reflecting surface of the transparent index mirror. S is any distant object or

heavenly body, the rays from which are reflected upward by MM.' By inclining MM' these reflected rays may be made to coincide with rays passing upward from PP'.

Consider the instrument held approximately level so that the bubble, moving freely to the highest point of its cell, will rest at a position B. The downward rays from B will be reflected in the penta-prism. Draw OPB, cutting the penta reflector at P. The ray from B incident at P will be reflected up through the point F, and on through the transparent mirror at M to the eye at E.

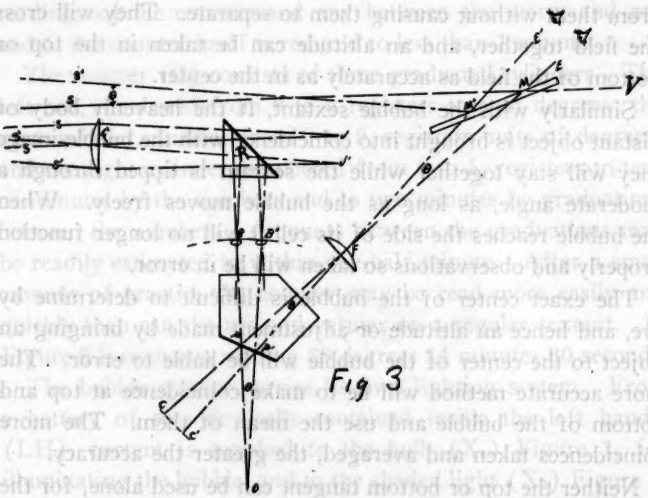


Fig 3

With the sextant held stationary, the index mirror M is now inclined until the ray SM is reflected so as to coincide with the ray BPFM along ME. Prolong SM to V, and it will be seen that MV is equivalent to a reflection of PFM as well as a continuation of SM. Considering BPFMV then as a continuous ray, we see that it obeys the law that when a ray undergoes two reflections in the same plane the angle between its first and last direction is equal to twice the angle between the reflecting surfaces.

But gravity acting on the liquid in the bubble cell makes the line BPO a true vertical between zenith and nadir. Therefore it follows that the angle between ray SM and ray BPO is the zenith distance of the body S, and is measured by half the angle between the index mirror and the penta-prism mirror. The latter is fixed with relation to the frame of the instrument; therefore the inclination of the index mirror, when measured by a suitable scale, can be made to record the true altitude of S directly.

Suppose now that the instrument be inclined through a small angle so that the bubble rests at B.′ The distant body is now at S′ relative to the instrument, but the angle between S′M′ and B′P′O has not changed from that between SM and BPO, and the same position of the index mirror still measures the altitude of A. Or putting it in plain language, the bubble moves through the field the same angular distance that the instrument is inclined, and hence its coincidence with the heavenly body is not affected by movement from B to B′, as the sextant is inclined.

In the foregoing demonstration only the single ray passing from B through F has been considered, for the sake of clearness. In actual fact an infinite number of rays radiate from any illuminated point. If the point be distant like S then the rays are received at the eye or the instrument sensibly parallel to each other. If the point be near like B, the rays are divergent but may be collected into parallelism by a suitable lens.

Referring to Figure 4 such a lens is placed at F, its focal center being conjugate to the point O, and its focal length being equal to the radius of the bubble cell and to FPB. It will be seen that all the downward rays from the bubble, B, are reflected by the penta-prism to the lense and emerge therefrom parallel to each other and to the central ray drawn through the focal center of the lens at F, as though B were at an infinite distance. In Figure 4 the heavenly body, S, is shown at an altitude of 45 degrees, so the mirror, M, must be inclined to

22½ degrees with reflector P to make coincidence between the bubble rays and those from S. Movement of the eye to different positions of E does not affect the coincidence.

Up to this point we have neglected the prism R and lens F′, located above and to the front, which play no part in the ordinary use of the sextant with the bubble, but function in the adjustment of the instrument and in using it on the sea horizon like the ordinary nautical type.

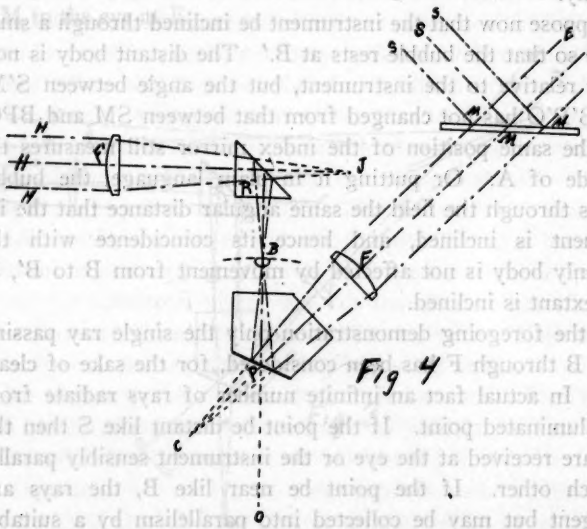


Fig 4

The lens F′ is so located with reference to the right-angled prism R and the bubble-cell that the distance F′RB equals its focal length F′RJ, and the bubble is therefore in the focal plane of both lenses.

Referring again to Figure 3, rays from S incident on the lens F′ comes to an imaginary focus at J and are actually reflected down by the prism R to the bubble-cell, which is transparent. By proper mounting of the prism, R, these rays may be made to focus at B. This adjustment is permanent and

thereafter if the instrument be tilted through θ, the rays from S′ follow the bubble from B to B′, with the result that an image of S may be seen in the field of the telescope, coincident with the bubble image.

The telescopic image of S is distinct from the image of S reflected in the index mirror M, but may be made to coincide with it by inclination of the index mirror, just as is done when finding the index correction of a nautical sextant. For simplicity we shall call the former the projected image of S, and the latter the reflected image.

To use this instrument as a nautical sextant, the sea horizon is kept in the telescopic field and the image of the sun or star, reflected in the index mirror, is made to coincide with the image of the horizon in the telescope, disregarding the bubble entirely.

The adjustment of this bubble sextant for either type of use is the same in general principle as that of an ordinary sextant; that is, a known angle, preferably zero, is measured, and the scale is either adjusted to read zero on this angle, or the difference from zero is carried as a constant index correction.

In Figure 3 assume S or S′ to be in the true horizon or actually level with the instrument, and that by turning the index mirror its reflected image in M and M′ has been brought to coincide with the bubble image.

The scale should now read zero degrees, zero minutes. If it does not, we may carry the difference from zero as an index correction, or we may eliminate it by resetting the scale to zero for this same position of the index mirror.

Similarly we may make coincidence between the image of S reflected in M and the projected image of S seen *through* M. If the index correction by both methods is the same, then resetting the scale to zero will eliminate it for both of them. If it is not the same then it means that the projected image of S does not exactly coincide with the bubble image, and a resultant error is bound to be left in one of them.

The following procedure for adjustment is recommended: To establish a true horizon range, set up a stand for the sextant

at some convenient point ashore; then with a theodolite or surveyor's level at the same height as the sextant sweep the skyline until some roof, chimney or other distinct point or line is discovered at exactly the same level and preferably not less than two miles away. It may be necessary to vary the height of the sextant stand to get this level accurate to less than a minute of arc. Call the point or line selected the true horizon point, H.

With its scale set at zero, set the sextant on the stand and swing it in azimuth until the two inverted images of the true horizon point are together in the center of the field laterally. Incline the sextant until the bubble is seen in the center of the field as nearly as can be determined by eye, and clamp or block it in that position.

Now disregarding the projected image, turn the index mirror by the thumb knob, K, until the reflected image of the true horizon point, H, is in the center of the bubble image. Since the center is difficult to estimate with accuracy, it is preferable to bring the true horizon point alternately into coincidence with top and bottom edges of the bubble for several readings of the counter which should be recorded. The mean of these readings will equal the reading for the center of the bubble, and should be zero, but may differ slightly; a mean reading of 99 degrees 58 minutes, for example, would be an index correction of plus 2 minutes on the bubble. (The scale is graduated from zero to 100.)

Now bring the same reflected image of the true horizon point into coincidence with its own projected image several times, and take the mean scale reading as before. If it came to 99 degrees 58 minutes also, it would show that the true horizon image centered accurately with the bubble image, and that the instrument as a whole read 2 minutes low, whether used with the bubble or with a visible true horizon. It would then only be necessary to shift the counter scale to read 2 minutes higher, or zero for the mean of several observations, or to carry the constant index correction of plus 2 minutes.

If the mean of the coincidences taken with the bubble did *not* agree with those taken on the projected true horizon point, it would indicate that the latter was not being projected through the exact center of the bubble. This can only be corrected by shifting the right angle prism (R), and it is not recommended that this be attempted except by an experienced optical mechanic.

If this error is found, it does not prohibit the use of the instrument in either way, for one may either carry in mind the two different index corrections, or the counter may be adjusted so as to take out the index correction on one, and leave it on the other. Thus, if the mean of the observations using the bubble were 99 degrees 58 minutes and the mean using the projected image of the true horizon point were 00 degrees 1 minute, we could leave them as index corrections of plus 2 minutes and minus 1 minute, respectively, or we could take it out of the former and have minus 3 minutes left on the latter.

The counter scale may be adjusted by slacking up the set-screws in the clamp collar, Q, on the end of the counter shaft, and turning the collar with the fingers until it reads zero. Be careful not to move the thumb wheel or counter shaft while doing so. In the zero position one set screw may be inside out of reach; turn this outside and slack it first before slacking the other, and the same when tightening up again.

Suppose the mean of several coincidences between the true horizon point and the bubble is 0 degree 03 minute, or an I. C. of -3 minutes. Turn the thumb wheel (and index mirror) until the scale reads 0 degree 03 minute, slack off the set screws carefully and turn the counter until it reads 0 degree 00 minute then tighten the set screws and recheck.

The index error found by coincidences between the reflected and projected images of H may be taken out in the same way, though it is presumably better to take it out of the bubble coincidence and leave the residual error, if any, in the use of the instrument as an ordinary nautical sextant.

The design of the instrument—light transmission shafting, small worm and gearing, and an index mirror supported only by light trunnion bearings—makes this instrument considerably more subject to change in the index correction than the ordinary nautical sextant, which has its index mirror mounted solidly to the index bar, and which reads directly from a graduated arc instead of through a gear-driven counter. It is, therefore, advisable to check the index correction before and after all observations.

But it is, of course, impossible to check the correction on the bubble without the aid of a true horizon point or distant body of known true altitude which will be impossible from a plane; whereas the correction for coincidence between reflected and projected images of distant objects may be taken at any time, using the sea horizon, heavenly body, or distant object.

Now if there is no difference between the two I. C.'s, or if the difference be known and assumed constant, then one I. C. can readily be found from the other. These assumptions are likely to be correct, for any slight displacement of the index mirror or counter gear will throw an equal error into both methods of using the sextant, and the two I. C.'s will not differ unless the prisms be displaced, which is not so likely to occur provided the instrument receives reasonable care.

The following method is recommended for handling the corrections: Using a true horizon point H, make coincidence between it and the bubble and get the index correction, calling it "B." Then make coincidences between the reflected and projected images of H and get the other index correction, calling it "D."

Consider this "B" − "D" as a constant to be recorded. Then subsequently, when "D" is observed but "B" cannot be, we can nevertheless find it by the simple formula "(B − D) plus D equals B," just as in taking the time of sights we add (C − W) to W to get C again. If no change is found in D, we may assume the same of B, while if D has changed B will have presumably changed an equal amount.

There seems no reason why a true horizon range could not be found or constructed at any Naval Station. All that is necessary is a stand for the sextant (possibly under cover), blocks or clamps for holding it in position, and a clearly defined point about two miles away and on the same true level as the sextant to within a minute of arc. A point made in the shape of a cross of timber would be excellent.

The following methods are suggested for testing on board ship; which, while rough, may at least be used as a check on previous adjustment:

(*a*) Using another ship for true horizon point. All that is necessary is that she be far enough away, that you can determine the height of some line or point aboard of her and set up sextant at the same height above waterline on your own vessel, and that there be not enough motion to change these heights. An error in relative height of one foot will cause an error in checking of only about 1 minute in a thousand yards, and less in proportion at greater distances.

(*b*) Using the sea horizon. Find the exact "dip" of the height at which you are working. Bring the reflected image of the sea horizon to coincide with the bubble image for several observations at top and bottom, and take the mean of them. The result, which will be below zero, or "Off the arc," should be the same as the "dip," or the difference between them is the index correction. Compare this with the index correction obtained by direct coincidence between the reflected and projected images of the horizon.

(*c*) By observations on a heavenly body. Select a certain watch time for the test ahead of time, reduce this to G.M.T., and calculate the altitude for this time just as is done in an ordinary line-of-position sight, correcting it for Semi-Diameter and P & R with reversed signs. Then take observations with the bubble sextant, starting a few minutes before the selected watch time, and getting as many as possible, recording the times of each. Plot a fair curve of times and altitudes, which will give a more accurate check than a single observation taken

at the exact instant selected. Any error in the elements used to compute the true altitude will, of course, affect the accuracy of the comparison.

(*d*) Comparison with an ordinary sextant. Take simultaneous altitudes by bubble sextant and by a good sextant of ordinary type. Correct latter for dip and any I. C. it may have. Plot fair curves of time and altitude for each and compare them.

Methods (C) and (D) would be less practicable for making an actual adjustment, but would be interesting as showing the performance of the instrument in measuring angles other than zero.

There seems no way of checking the index correction on the bubble from a plane in flight, for the "dip" would presumably be too inaccurately known to use the sea horizon, and any form of true horizon is out of the question. The index correction by direct coincidence may be taken on the sea horizon, however, and if it has not changed, it is safe to assume that the index correction on the bubble has not changed either, or if it has changed, they have both presumably changed alike, since the change is more likely due to the mirror or counter than to the prisms or lenses.

Some other points to be noted in connection with this instrument are as follows:

There may be a certain amount of lost motion in the worm and counter gear, and this should be ascertained by a series of observations on the same point, moving first from high to low on the scale and then from low to high, and comparing the two averages. To eliminate this error it is desirable to make all adjustments and observations moving in the same direction.

In taking observations of stars or planets the instrument is held *above* the eye instead of below. The bubble image is then seen reflected from the lower side of the index mirror instead of through it, and the star is seen *through* it instead of reflected. No change in principle is involved. It is so used

because the star, being a difficult target to locate and hold at best, can be seen better directly through the mirror than when reflected from it.

Considerably more practice is required with this instrument than with an ordinary nautical sextant, as it has to be held fairly steady with reference to three different planes, or axes. Tilting it in the plane towards or away from the target causes the images of bubble and target to cross the field vertically together until the bubble touches the sides of its cell, after which it no longer functions.

Swinging it in azimuth does not move the bubble, but causes the inverted image of an object projected through the telescope to cross the field in the same direction, while the inverted image of the object reflected in the index mirror crosses the field in the opposite direction.

Tilting it sidewise causes the bubble to cross the field laterally, as well as tilting the image of any body reflected or projected. This throws in the same error as occurs in the nautical sextant when the sun, instead of being brought straight down to the nearest point of the horizon, is brought down obliquely.

The best way to observe is (1) Set the scale for the approximate altitude of the body to be observed, (2) hold sextant so that bubble appears approximately in center of field, (3) swing it in azimuth until sun or star is vertically in line with the bubble, and (4) turn thumb knob to make coincidence.

On first looking into the field and noting the images of the bubble and distant body, there appears to be a large amount of parallax between them. On closer examination it will be seen that the parallax is in the bright center, instead of the bubble as a whole, due to the fact that the bubble itself becomes a sort of irregular lens. All estimates of the center of the bubble should, therefore, be made with reference to the outer circumference of the bubble, instead of the inner part.

There is some true parallax at the sides of the field, so that observations should be taken at the center as far as possible.

The great difficulty in taking accurate observations is due to the vibration or swaying of the bubble due to unsteadiness of the hand, as distinguished from its natural movement under the force of gravity and inclination of the instrument. When used from a plane or from a ship in a seaway, it will probably be best to bring the sun or star to the center of the bubble, taking several observations as rapidly as possible and using the mean of them.

For adjustment, when the instrument can be held steady, it will be more accurate to take the mean of coincidences with the top and bottom of it, as before stated.

DISCUSSION.

LIEUTENANT M. F. SCHOEFFEL, U. S. NAVY.

The aircraft sextant is necessarily different from the Marine sextant, for, when the plane is at any considerable altitude the horizon is generally not visible. Even in case the horizon can be seen the altimeter does not indicate the altitude with sufficient reliability to determine the height of eye correction accurately.

For these reasons the aircraft sextant must embody its own horizon within itself. Three lines of approach to this desideratum have been attempted:

(*a*) Gyroscopic horizons.

(*b*) Pendulous horizons.

(*c*) Bubble horizons.

The gyroscopic sextants have been either mechanically or operationally complicated. The pendulous sextants have found little favor. Development has been mainly concentrated upon bubble sextants, because of their simplicity.

All three of these types of sextants suffer from the disadvantage that the horizon element is acted upon, not by the gravitational acceleration alone, but by the resultant of the gravitational acceleration and whatever other accelerations may

exist upon the airplane at the instant of taking the sight. No device suitable for incorporation in an aircraft sextant is known at the present time which will react to the gravitational acceleration alone, ignoring the other accelerations.

Since these accelerational errors, which frequently are of the order of 2 degrees, or 120 miles, are inherent in the design of the sextant, it follows that success in its use is very largely dependent upon the skill of the pilot in flying the plane steadily. The best pilot cannot fly with even approximate steadiness in bumpy air, and even in smooth air the steadiness of flight is by no means absolute.

Experience has shown that the best results can only be obtained by taking a number of sights (six or seven) as rapidly as possible and averaging the readings. In still air, an error of seven or eight miles may be expected when this method is used, and in bumpy air an error of twenty miles is good work.

Since the major source of error in aircraft sextants, as built up to the present time, is due to the inherent defect of the physical principle involved, the designers have concentrated largely upon attempting to develop an instrument which is handy to use. Handiness is considered to be essential in aircraft sextants because of the difficulties imposed by cramped space, plane vibration, and the fatigue incidental to flying.

The Type A sextant described by Commander Parker is considered to be a good example of a modern instrument designed with these difficulties in view. It does not overcome the fundamental accelerational error.

As yet no satisfactory solution of the problem of the accelerational error has been devised and the aircraft sextant remains an inaccurate instrument when used in the air.

ABSTRACTS FROM THE GERMAN TECHNICAL PRESS.

By E. C. MAGDEBURGER, MEMBER.

THE APPLICATION OF CIRCULATION THEORY TO THE DESIGN OF HYDRAULIC PROPELLERS—RESUME OF PAPER READ BEFORE THE GERMAN SHIPBUILDERS SOCIETY ON 20 NOVEMBER, 1925, BY DR. PHIL. HERMANN BORCK. "WERFT-REEDEREI-HAFEN," 7 DECEMBER, 1925, P. 716.

The circulation theory originally applied by Prandtl, Bendemann, and their associates to aircraft propellers has gradually found application in the design of hydraulic propellers. It is based on the assumption that phenomena similar to those occurring around the wings of aircraft develop also about the propeller blade on its helical path. The water is accelerated by the propeller and three different velocities—before, around and after the propeller, with corresponding stream cross-sections—are differentiated. Accelerations develop in the axial direction, radially (stream-contraction) and tangentially. The circulation theory binds all these characteristics of the motion with the diameter and thrust and also with horsepower and efficiency, so that by means of force = mass $\times$ acceleration and work = $m\ v^2$ a performance analysis can be made as well as limits of propeller efficiency established.

The author developed the fundamental efficiency formulas and demonstrated the determination of the two components of the total propeller efficiency by means of curves. A practical application was shown on an example.

Summarizing the author mentioned especially the value of the circulation theory when determining limiting figures for the efficiency of a propeller.

DISCUSSION.—Dr. Schmidt (Berlin) described a more accurate method of determining the width of the blade based on curves, embodying the results of his own experiments. He also stated that the trials of the American torpedo boats show that cavitation by ship propellers begins when the product of the revolutions and the pitch reaches about 16 meters per second (52.5 feet per second), which can be explained by the fact that a water drop falling from a height H = 33.9 feet, corresponding to a height of the water column for atmospheric pressure, acquires a velocity $v = \sqrt{2gH} = 14.2$ meters per second = 46.6 feet per second.

Dr. Bauer, Hamburg, doubted the possibility of calculating the most efficient propeller analytically, particularly for special ships. The most important factors influencing the efficiency could be obtained by trial only.

NEW COMBINED DYNAMIC-STATIC BALANCING MACHINES.

RESUME OF PAPER READ BEFORE THE GERMAN SHIPBUILDERS SOCIETY ON 20 NOVEMBER, 1925, BY DR. HORT. "SCHIFFBAU," 9 DECEMBER, 1925, P. 757.

New fundamentals of balancing of rotating bodies have been developed by Krupps when failures with purely dynamic balancing made static balancing appear essential. The static unbalance is responsible for essentially different vibration than that caused by dynamic unbalance. Effects of static unbalance appear at lower rotating speeds, whereas those of the dynamic unbalance frequently do not appear at all since the necessary high speeds are often above the operating range. Several different models of balancing machines were described. The shaft with the body to be balanced is simply laid down on the roller bearings of the balancing machine, which can be adjusted to suit a wide variety of work. The order of balancing is immaterial. To change from one to the other the dynamic

balancing head with springs only must be added or taken off. Both parts of the method were then described in detail. The dynamic unbalance is being corrected by moving the rotating masses of the balancing head, which correction manifests itself in the disappearance of the critical vibration. Thus the process requires that the body under test be rotated in or around the critical speed only once without interruptions. The unbalance thus obtained by means of weights on the balancing head can be readily corrected by the use of tables. This simple method is claimed to accelerate the balancing work very materially.

Among the advantages claimed are: (*a*) mounting of work by simply laying down on the roller bearings; (*b*) no limits to the dimensions of the work to be balanced; (*c*) reduction of the time for dynamic balancing to a half of the usually required; (*d*) gradual speeding up into the critical without the necessity of excessive speeds; (*e*) convenient interchangeability of the spring resistance; (*f*) shortening of the dynamic balancing through the balancing head.

The author pointed to the necessity of a balancing method for balancing rotating parts when installed in place and in the housing, rotating at operating speed. Fundamentals of such a method were given and further details promised to be published in the near future.

DISCUSSION.—Eng. Schenk was of the opinion that the combined method is not suitable for speeds exceeding the critical speed, when elastic deformations will destroy the balance and maintained that the dynamic balancing machine of Heymann-Lawaczek, though more difficult to manipulate, still does not require as much time as the method described. Prof. Weber was of the opinion that the Heymann-Lawaczek balancing machine insured static balance also. In conclusion the author admitted that the dynamic method is not always unsatisfactory but quite frequently so and pointed out to American practice as confirming this opinion.

BENKERT ATOMIZER FOR HIGH SPEED OIL ENGINES. BY JULIUS EGGERS. "SCHIFFBAU," 23 DECEMBER, 1925, P. 799.

The superiority of the oil engine is now well recognized and the field of its usefulness continuously expanded in all parts of the world. Of especial interest are the efforts to adapt it for high speeds and small powers required by ship's auxiliaries, motor boats and by the automotive industry. Motors of usually 4-cycle type with airless injection of fuel either direct or through an ignition chamber have been already developed. Reliability of such a plant is, next to fuel economy, of the most importance and reliable operation as well as readiness to start are now the goal of the engine designer. With this in mind, the present type of 4-cycle airless injection oil engine has still too many delicate, attention requiring parts and its modification so as to include also the ability to burn all the commercially available low grade fuel oils and coal distillates reliably and economically appears very desirable. At present these motors cannot do so without much difficulty. As far as known to the author there are no 2-cycle engines of high speed type on the market that can start reliably on heavy oil without outside ignition, unless these are burdened by other undesirable complications, such as excessive compression, etc.

The best fuel consumption of the 4-cycle engine is reported to be .365, rising to .40 pound per B.H.P. per hour under normal operating conditions for the direct injection type and .44 pounds for engines with the ignition chamber. Two-cycle engines though distinctly superior as far as reliability of operation is concerned, use as a rule some 8-10 per cent more fuel due to inferior combustion process, which cannot, however, be considered of great importance especially since those of the direct injection type, when equipped with the Benkert atomizer have as favorable a fuel consumption as the 4-cycle with ignition chamber. Besides the existing 4-cycle engines

are not capable of burning low grade fuel. Thus a 2-cycle engine with Benkert atomizer for direct injection, without ignition chamber and outside ignition is far superior to any existing high speed engines with airless injection.

Figure 1 represents schematically the principle of the Benkert atomizer. The fuel entering canal *a* under a pressure of only 750 pounds per square inch, is divided into two separate streams *b* which strike each other in canal *c* with a velocity of about 400-500 feet per second. As a result even the heaviest fuels are thoroughly atomized and broken up. From the central point of impact one or more openings *d* of about .7 mm.

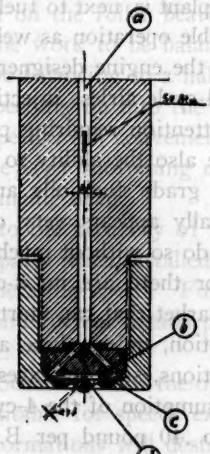


FIG. 1.

(.028″) lead under suitable angle into the interior of the cylinder. These discharge the fuel in so finely an atomized state that it ignites even with moderate compression pressures without the assistance of outside ignition.

One of the types of atomizer at present used is an open nozzle type with four holes of .008—.010 inch requiring

a pump pressure of 3500-4300 pounds per square inch for satisfactory atomization and a similar pressure has to be used with another atomizer of the closed type with spring controlled needle.

Photographic results of comparative tests of the three atomizers are given in the original article together with a drawing showing application of the Benkert atomizer to any existing 2-cycle engine with an ignition chamber where this atomizer permits cold starting without outside ignition, such as electric wires, cartridges or torches.

The Influence of Shape of Cross-Section for the Pressure Hull of a Submarine Upon Stability When Diving and Submerged. By M. Harun, "Werft-Reederei-Hafen," 22 January, 1926.

Much mystery has surrounded all progress in the development of the submarine giving rise ofttimes to rumors of exceedingly exaggerated claims. Hence a scientific investigation of the basic differences between the ordinary cylindrical shape of cross-section of a submarine pressure hull and that proposed and patented by a noted German submarine expert, Prof. Flamm, is particularly welcome. The article deals exclusively with the influence of the shape of cross-section of the pressure hull upon the stability of a double hull submarine vessel both during diving and while running submerged, leaving completely out of consideration questions of relative strength.

In order to further narrow down the scope of the article the author considers only the tri-circular cross-section based on an equilateral triangle, with the center of the lower circle in the center of gravity of the triangle while the centers of the side circles are located on the perpendiculars out of the middle

of the other two sides, thus giving it a characteristic shape shown on the accompanying Figure 1.

Neglecting the trimming down of the shape toward the ends the author considers only a unit section of equal cross-section and even the outer hull is ignored in order to get conclusions of a more general character.

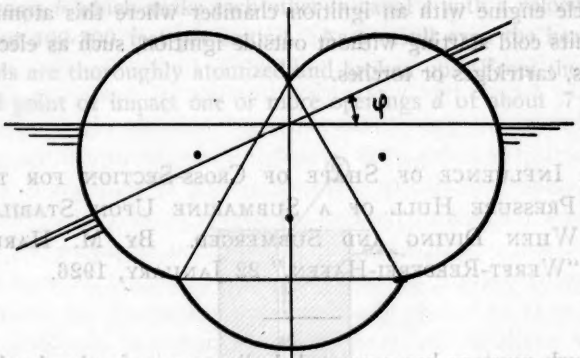


FIG. 1.

The influence of the distribution of weight for similar reasons was not considered, though this factor can be easily accounted for in the final results.

A mathematical investigation of the plain cylindrical cross-section leads to the conclusion that the leverage of the static moment of form stability is a harmonic function, i. e., it is proportional to the sine of the angle of inclination φ, whereas similar mathematical treatment of the tri-circular cross-section leads to very complicated expressions for the moment of form stability. A graphical interpolation method, described in detail, is used for the purpose of evaluating these formulas.

The curves resulting from such a method of investigation are claimed to show clearly and beyond doubt that the initial stability for tri-circular cross-section is quite considerably

larger than for the plain cylindrical. Furthermore it is demonstrated that for the angles of inclination of practical value the moments of form stability grow quicker and larger for tri-circular section than for the plain circular, for which the maximum moment is always reached by 90 degrees, whereas for the former the maximum depends on the arrangement of the side circles and the angle of inclination and may be reached by from 50 degrees to 75 degrees angle of inclination.

Hence the claim of the author that the tri-circular cross-section for submarines tends to increase materially the safety of the vessel as compared with the plain cylindrical shape of pressure hull.

SOLUTION OF STATIC PROBLEMS BY MEANS OF MODELS. By OTTO GOTTSCHALK, "V.D.I.," 20 FEBRUARY, 1926, p. 261.

Every problem in statics can be reduced to the form $R = Py$ wherein P is some given external force, R is the resistance of any given section of the loaded beam, such as reaction of supports, bending moment or shearing force, and y denotes the inner elastic deformation, which can be simulated on a suitable model.

"Continostat Gottschalk" permits the simple representation of the beams and their loading in such a way that they build influence lines and thus serves for direct reading of resistance. Furthermore it reproduces directly and reliably the elastic deformation under any given loading and thus permits the simple evaluation of occurring stresses.

The apparatus consists essentially of a flat metallic beam of heavy cross-section on which can slide supports, through the upper ends of which passes the elastic beam of the model. Flexible steel beams of different length and thickness permit reproduction of any given construction, with numerous smaller

pieces serving to express the influence of local changes in cross-section in any part of the structure. A beam with clamps on it permits a frictionless and yet restricted relative motion of various elements, such as for bases of columns with respect to beams as in portal or frame constructions. Angle clamps serve to reproduce curved systems and also break in the beam for the direct reading of bending moments—clamp 1—for 90 degrees connection, clamp 2 —for bending moments within the span, clamp 3—for half of the support moment for straight line beams, clamp 4—for half angular moment of two perpendicular beams. Theoretical basis for the method is well covered in the current technical literature by the author and others.*

The purpose of the present article is to call attention to the importance of mechanical calculations which in the future will reduce the work on problems of statics to a very small fraction of that now required, completely eliminating the sources of errors unavoidable with the usual lengthy purely mathematical computations.

The original article contains ten examples showing how apparently complicated systems can be quickly and simply resolved by means of the apparatus developed by the author without much drawing or calculation work and automatically for any kind of loading conditions. A further example illustrates an application of the method to the determination of moments in any member of a truss under a given load by tracing the shape of the actual elastic deformation curve, which can be done simply and accurately and without calculation by means of the apparatus described.

Example 1 is illustrated and described below:

A continuous beam of three spans—5m, 9m and 6m is loaded in the first span by a distributed load $G_1=2\times1=2t$, in the second span by a distributed load $G_2=9\times.5=4.5t$ and a

*(a) D. B. Steinman—Influence lines as deflection diagrams—Eng. News—Record, Vol. 74—(1916)—p. 648; (b) George Erle Beggs—Proc. Am. Concrete Inst.—13 Feb., 1922. Accurate Mechanical Solution of Statically Indeterminate Structures by use of paper models and special gauges; (c) O. Gottschalk in various South American technical publications in Spanish.

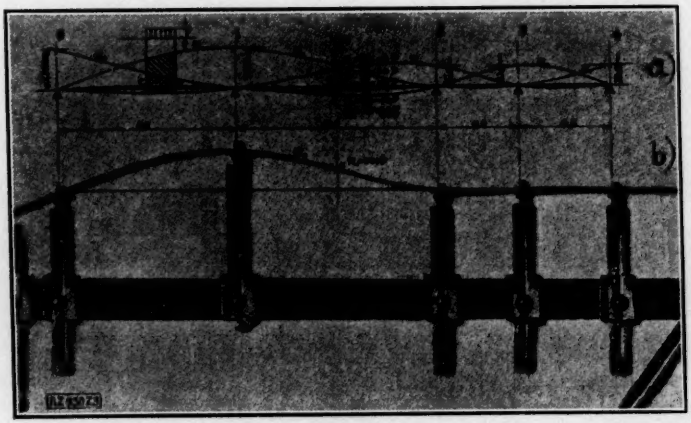


FIG. 2.

concentrated load P=6*t*. Reactions R_0 and R_1 are to be determined. A slide is fastened in place of each support on the

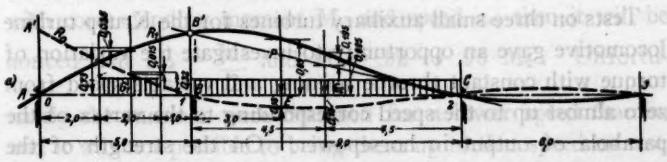


FIG. 1.

drawing. The slide over support B, the reaction of which is to be determined, is raised or lowered any given distance (but not more than $\frac{AB}{5}$) up to B' and the resulting curved line shown in full on Figure 1 is the influence line for R_B, the ordinates of which all have to be divided by the distance BB'. If BB' = 3 cm. then the ordinate under the load P shown as 2.67 cm. by the influence line becomes $\frac{2.67}{3} = .89$.

With ordinates as shown on Figure 1 the reaction R_B becomes

$$R_B = G_1\left(.73 - \frac{.026}{3}\right) + G_2\left(.695 - \frac{.195}{3}\right) + P \times .89$$

$$= 2 \times .72 + 4.5 \times .63 + 6 \times .89 = 9.62t.$$

In a similar manner the slide at A is raised 2 or 3 cm. up to A' resulting in the influence line for R_0, shown by a dotted line with ordinates already divided by AA' also given on Figure 1. Hence

$$R_A = 2\left(.32 - \frac{2 \times .015}{3}\right) - 4.5 \times \frac{2 \times .18}{3} - 6 \times .18 = 1.0t$$

A further illustration of similar problem is given on Figure 2. It is quite natural, of course, that the influence lines thus obtained can be used for reading off the ordinates for any other kind of loading.

STEAM TURBINES WITH HIGHLY VARIABLE SPEED. BY DR. RUDOLF LORENZ, "V.D.I.," 6 MARCH, 1926, P. 314.

Tests on three small auxiliary turbines for the Krupp turbine locomotive gave an opportunity to investigate the variation of torque with constant throttle opening and variable speed from zero almost up to the speed corresponding to the vertex of the parabola of output in horsepower. On the strength of the results of these tests a simple method to determine horsepower, torque and efficiency for all speeds is suggested.

The test results show that the torque M—Figure 1, with constant throttle opening is a linear function of rotative speed (R.P.M.) and only at very slow speeds does this relation

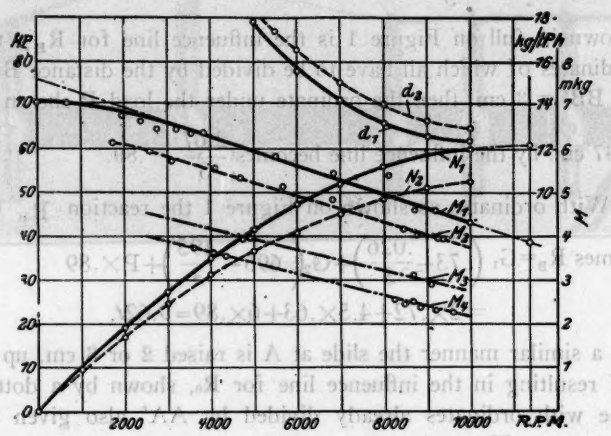


FIG. 1.

N_1 H.P. Output at $p_1 = 12$ at. abs. $p_2 = 1.2$ at. abs. $t_1 = 300$ °C.
N_2 H.P. Output at $p_1 = 11$ at. abs. $p_2 = 1.2$ at. abs. $t_1 = 300$ °C.
M_1 Torque at $p_1 = 12$ at. abs. $p_2 = 1.2$ at. abs. $t_1 = 300$ °C.
M_2 Torque at $p_1 = 11$ at. abs. $p_2 = 1.2$ at. abs. $t_1 = 300$ °C.
M_3 Torque at $p_1 = 8$ at. abs. $p_2 = 0.9$ at. abs. $t_1 = 286$ °C.
M_4 Torque at $p_1 = 8$ at. abs. $p_2 = 1.2$ at. abs. $t_1 = 278$ °C.
d_1 Steam consumption, $p_1 = 12$ at. abs. $p_2 = 1.2$ at. abs. $t_1 = 300$ °C.

change. If this line be continued to the ordinate axis and the moment of the intersection be designated by M_o', if on the other hand the horsepower curve be exterpolated to its vertex with corresponding moment M_s and speed n_s, then it will be noticed that $M_s = \frac{M_o}{2}$ and $M_o = .92$ to $.96\ M_o'$. Unfortunately the turbines were designed for a lower speed than n_s and it was not possible to bring them up to that speed on test, although turbine No. 3 came sufficiently close to it to make the above deduction unimpeachable.

This leads to the following simple method for the determination of horsepower of turbines working with constant quantity of steam and highly variable speed, such as the locomotive drive turbines, which obviates the long thermodynamic calculations for each given speed and the results of which are sufficiently accurate for a number of purposes.

The turbine is first calculated thermodynamically for the speed n_s at the maximum horsepower (at vertex of parabola). From the speed and horsepower the torque M_s is determined and from that the theoretical starting torque $M_o' = 2M_s$. These two points are connected by a straight line (Figure 2).

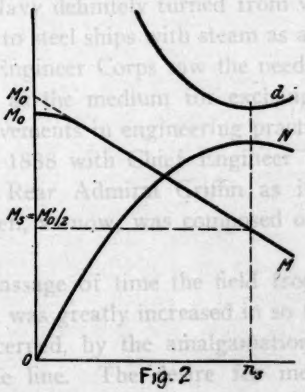


Fig. 2

The actual moment line can be approximated by assuming $M_o = .92\ M_o$ and drawing from that point a slightly curved line, which will join the straight line at a speed of about $\frac{1}{4}\ n_s$. Thus the horsepower can be determined for any speed and the curve N plotted. The steam consumption d can also be found by dividing the total quantity of steam for which the calculation is being made G by horsepower output N, so that $d = \frac{G}{N}$ in kg per H.P. per hour. Both the horsepower and steam consumption curves up to $n = \frac{1}{4}\ n_s$ are of exactly parabolic shape.

The tests also showed that for partial openings the torque lines are also straight and that the relation $M_o' = 2M_s$ remains valid if M_s is referred to the vertex of the new horsepower curve and opening. The vertices move toward the origin, however, and in order to determine the output with partial openings the turbine must be analyzed thermodynamically for two different speeds. Thus two points of the horsepower curve are obtained and permit the determination of the corresponding torque values and the drawing up of the torque lines and horsepower curves throughout the whole range of speed.

THE ANNUAL BANQUET.

The annual banquet of the Society was held at the Willard Hotel, Washington, D. C., on March 27, 1926. More than one hundred and sixty members of the Society and their guests attended the dinner, which was pronounced a very successful affair. Among those present were five past Presidents of the Society, including Rear Admiral G. W. Baird, at whose instigation the Society was formed in 1888, and Rear Admiral R. S. Griffin, the first Secretary of the Society. Rear Admiral M. M. Taylor presided, and Commander Miles A. Libbey ably fulfilled the duties of toastmaster.

Admiral Taylor introduced Commander Libbey with the following remarks:

"Guests and fellow members of the Society: It gives me great pleasure to welcome you to this dinner which serves to commemorate the 38th year of the Society's existence.

"When the Navy definitely turned from wooden ships with auxiliary steam to steel ships with steam as a prime mover, the officers of the Engineer Corps saw the need for some organization to serve as the medium for exchange of information as to the improvements in engineering practices. The Society was formed in 1888 with Chief Engineer Towne as its first President and Rear Admiral Griffin as its first Secretary. The Society then, as now, was composed of Naval and Civil members.

"With the passage of time the field from which members could be drawn was greatly increased in so far as Naval members were concerned, by the amalgamation of the Engineer Corps with the line. The desire for membership was increased by the great increase in the use of mechanical devices

aboard ship and the much greater number of officers who must have knowledge of such installations. The close relation between ship practice and shore practice, the growth of our merchant marine and the development of shore installations for ship use, has led to the interest of our Civil Members and to their valued membership in the Society.

"The changes in the subjects dealt with during the life of the Society have been remarkable. When organized the best we had was the old *Atlanta* and *Boston,* 3500 tons and 16 knots; an electric plant which served only to light the ship. To-day we have the *Colorado* class, 32,000 tons, 21 knots, electric drive and sufficient electric output to light a good sized town.

"The Journal of the Society is a record of advancement from the simple steam installations of 1888 to the great turbine installations of to-day, to the electric drive, the vast increase in use of electric prime movers, the internal combustion engines for surface, sub-surface and air propulsion and the radio. It is a wonderful record of progress, and through the efforts of the members the steps in this progress are ably recorded in the pages of the Society's Journal.

"To me the remarkable element in this record is the wonderful increase in use of power aboard ship. When I was a midshipman on the old *Chicago,* we had an 8-inch gun rigged to train by motor. The control was poor and once started it frequently ran until it brought up on the stops and a fuse blew. Its main use was by the Captain who would start it to show guests he wished to impress with the modernity of this ship. To-day the 14-inch triple gun turrets are controlled within a fraction of a degree. The same advance is everywhere. Ships of to-day do everything by the opening of a valve or the throwing of a switch. But with this advance in the use of power units, the demands on the man have become greater; main strength has given place to intelligence. It is well stated by a British Admiral with a poetic bent in a poem he wrote, one verse of which goes like this:

'In this age of great invention we're instructed to believe
That the pressure of a button is the same as work achieved;
But the optimist inventor should remember if he can,
Though the instrument be perfect there are limits to the man.'

"As an evidence of the sustained interest in the Society, we have here to-night past Presidents whose efforts contributed so much to the success of the organization.

"We have with us a number of gentlemen distinguished in their careers who have kindly consented to speak to us to-night, and I will turn over to Commander Libbey the duty of toastmaster in order that he may introduce them and that I may sit at ease and listen."

Commander Libbey likened the dinner of the Society to a full power run—its annual chance to show its speed. Although, as he said, there is nothing in the Manual of Engineering Instructions regarding the operation of a Toastmaster, all hands think he is now qualified to write a chapter on this subject.

The first speaker of the evening was the Honorable Burton L. French who spoke on "The Navy and the Congress." He discussed the increase in Naval appropriations during the twenty years he had been in Congress and invited attention to the Budget system of balancing expenditures of the Government against its income. ". . . we do realize the importance of the Naval establishment. We want it to be great, we want it to be what you men have made it to mean, but at the same time we are compelled to look at all the great institutions of our Government and try in the mosaic to fit the Navy within so that it will be consistent with the great expenditures for every other branch of the Government service."

Rear Admiral Charles B. McVay, who has lately returned from command of the Yangste patrol, spoke entertainingly of the Navy in China. The Yangste river is patrolled for 1400 miles by seven American gunboats. The collection of raw materials and the distribution of imported goods to the

amount of $200,000,000 annually is made possible only through the efforts of these gunboats in keeping the river open.

Captain J. O. Richardson told most interestingly the "black and tan" story of an Englishman and a Chinese in the mixed court at Shanghai. Anyone who has been in the Far East would recognize the intonation even if he could not translate the harangue.

Dr. George Otis Smith, of the Geological Survey, spoke on "The Conservation of Natural Resources." The prodigal use of our coal, oil and metals with the rate of increase getting larger each year should make the engineer take thought for the morrow. Dr. Smith gave some interesting figures on the energy consumption of this country.

Rear Admiral Frederick C. Billard, of the Coast Guard, spoke on "The Viewpoint of the Coast Guard."

"I would invite attention of all who are interested in naval activities to the striking similarity between these operations of the Coast Guard and many factors of naval administration and operation in time of war. The problem involves a blockade, not of a certain section of the coast as was the case in our Civil War, but of 10,000 miles of American coast line; not a blockade as is usual in war to prevent enemy ingress into two or three ports, but a blockade to prevent unlawful entrance into any one of the innumerable harbors, bays, rivers and inlets that line so tremendous an extent of coast line, and to prevent landing on any spot of that coast line. It involves scouting operations with the sphere of activity of the enemy operations shifting suddenly from one spot to another. It means not concentrated activity for certain periods with intervals of repose when the enemy fleet is lying quietly at its base; it means constant activity day and night in all seasons and in all weather. Just as in actual warfare, care must be taken to preserve the rights of neutrals; meticulous attention must be given to navigation and to all other questions so that the seizure of an enemy ship may be made with a nice observation of all the technicalities of the law. A foreign vessel well

known to us as a notorious rum runner that has solemnly cleared from Halifax for Nassau laden with liquor, and that has been lying for six weeks off our Long Island coast, must be regarded, it would seem, as a perfectly innocent vessel engaged in a legitimate enterprise and manned by a number of estimable gentlemen, who in the course of their voyage to the West Indies have decided to delay for two or three months and enjoy the salubrious climate of our North Atlantic coast in winter time. It would never do to disturb those gentlemen as long as they are anchored a few feet beyond certain limits that no one is able to define with any authority. In time of war you send your orders to your vessels by secret code because the Navy has a highly elaborate communication system of its own. The enemy sends his orders by secret code. You try to pick up his message, to read his code; precisely that is an every day work of the Coast Guard to-day. In time of war the enemy maintains an elaborate espionage system. His spies hang around your bases, try to obtain information about the movements of your vessels, and try to undermine the morale of your men. The enemy maintains a system of propaganda that strives to weaken and embarrass your forces in the eyes of your own people. Just exactly those things are outstanding features of the warfare that the Coast Guard is now waging, without the freedom of action that war brings, but in a manner strikingly similar to many naval activities in time of war. What then is the viewpoint, the purpose of the Coast Guard to-day? To live up to its tradition, to its standards that have been stabilized for over a century and a quarter of honorable service; in accordance with those traditions to be a consistent and devoted ally and supporter of its great sister service the United States Navy, of which it becomes an integral part in time of national emergency; to render the utmost service to the people of this country, and particularly to those who go down to the sea in ships, and to add fresh laurels to its fine record of lives saved and property rescued at sea; to maintain a high state of efficiency with

respect to engineering; to steam ahead in line with the aspirations and ideals of this Society of Naval Engineers; and finally with respect to this great problem of smuggling work upon which I touched, to push ahead with determination and loyalty to a successful conclusion of the whole matter simply because it is a plain duty that has been laid at the feet of the service. The Coast Guard realizing the character of this task, its lack of popular approval, is looking for no alibi. It has already won pronounced success, but very much remains to be done, and the service is energetically, with determination, pushing on to mop up the deep. Whatever be your own thought as to this great social experiment in which the country is engaged, the officers and men of the Coast Guard at sea have given and are giving evidence of loyal, unselfish, efficient devotion to duty that it is the habit of all true blooded Americans to recognize and applaud."

The arrangements for the dinner were made by a committee consisting of Captain Cluverius, Commander Charlton, Lieutenant Commander Root, Lieutenant Commander Gillette, and Lieutenant Commander Cochrane.

DISCUSSION OF "THEORY OF THE IMPELLER," BY ALBERT E. GUY.

By E. F. Doty.*

Reference to the treatment of suitable ratio of $\frac{C_i}{C_o}$ which is highly important and fundamental in the design of a satisfactory impeller, it appears to this writer that the effect of inlet conditions on the characteristic have not been sufficiently stressed, possibly because these were not traced to their source.

On page 36 the author cites a case where one pump manufacturer claimed exclusive ability to design his apparatus so as to prevent overloading the prime mover. Presumably this reference is the so-called "Non-overloading" impeller which was in vogue several years ago.

The characteristic devloped by these impellers show a steep and sudden drop when operating under ordinary suction lift conditions, such for instance, as would obtain in ordinary shop testing.

The location on the characteristic when this drop takes place may be controlled by the designer at will and may take place at the point of maximum efficiency, or beyond that point. Therefore, such impellers are extremely sensitive to change in the suction conditions so that an increase in the suction lift will effect a drop in the characteristic earlier than with a smaller lift, while, by operating under conditions where a sufficient positive head on the inlet is maintained, the characteristic may be carried on to a normal one, and the drop will disappear. Under these latter conditions the "non-overloading" feature may disappear also.

* Assistant Chief Engineer, Buffalo Steam Pump Co.

The point in designing such impellers is the control of inlet velocity to produce the drop in the characteristic at the predetermined capacity which must be suited to the specific requirements.

However, this type of impeller has been largely discontinued as it is rather unsafe to use it for pumps where the inlet conditions are not well determined and there is no necessity for restricting the inlet area to the extent required when proper normal design may be utilized which will give the desired prevention of overload.

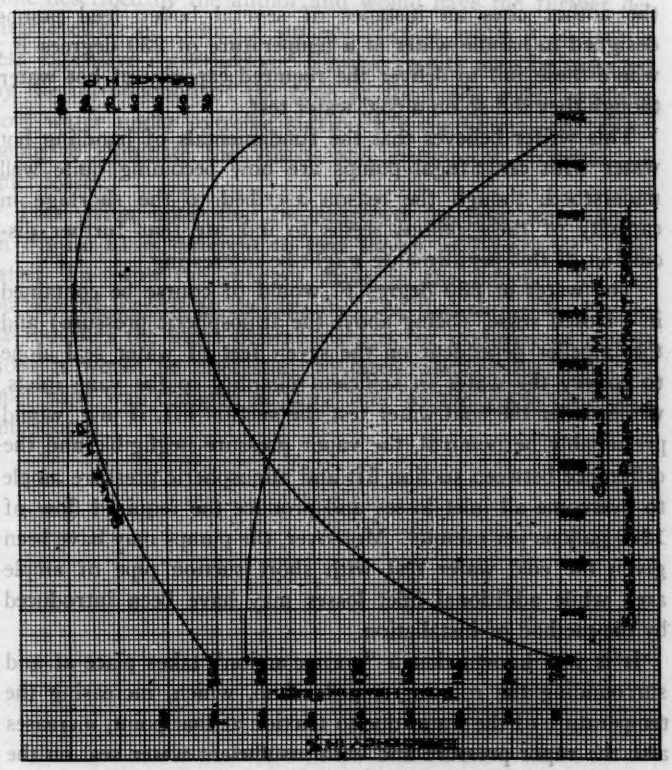
Control of the power characteristic may be effected by different methods of design but unless the actual operating conditions are known in each particular instance, there is but one method by which the pump characteristic may be predicted with a fair degree of certainty.

This method, which will cover the largest number of practical requirements, consists of the proportionate designing of the impeller outlet to control the power characteristic and compel a gradual drop in same, which is practically independent of the inlet conditions.

This writer is permitted, through the courtesy of the Buffalo Steam Pump Co. to present herewith a complete set of characteristic curves made from test of a commercial single stage pump operating at constant speed. It will be noticed from these curves that the power curve begins to drop gradually as soon as the point of maximum efficiency has been reached, while the "Head-Quantity Curve" is developing normally and without sudden fluctuation. This type of characteristic furnishes absolute safety against overloading the prime mover and stability of performance as regards the head and capacity over a wide range of suction and discharge conditions.

On page 43 the author states that for many years he has used the ratio $\frac{C_i}{C_o} = \frac{10}{7}$ but that on certain occasions some other ratio was used, better suited to meet special requirements. It seems to this writer that further explanation and

qualification would be advisable, as to when and how the ratio may be varied. It is quite probable that no definite rules can be laid down as the ratio may be varied to suit conditions at inlet, capacity and total head, but the tendency may be indicated.



For instance, in case of high suction lift or when handling high temperature water or handling water from a vessel under high vacuum and with small water seal; in the two latter instances and when delivering against high discharge head, in

either instance, the ratio may be easily reversed and become $\frac{C_i}{C_o} = \frac{7}{10}$ with resulting benefit to the pump efficiency and stability of performance.

On page 63, under sub-heading "Entrance into the Impeller," the author cites an instance of several boiler feed pumps designed to handle water at a temperature of 230 degrees F.; where these fail to deliver the required capacity of hot water though they did so on a cold water test.

This writer believes that the fundamentals of handling hot water with centrifugal pumps are now becoming quite well understood, though the reason ascribed to the shortage in capacity of these pumps seems to indicate that further discussion of this performance might be interesting. .

The water at 230 degres F. would of course be contained in a closed vessel (being above the atmospheric pressure) and the impeller dependent on the seven feet of water seal alone to convey the water in sufficient quantity to the vane inlets, while under the cold water test the suction lift, if any, would presumably be nominal, the vapor pressure negligible and the difference between suction lift and atmospheric pressure ample to overcome all inlet losses and produce the required flow of 1200 gallons per minute. Moreover, the pumps may have been given the cold wafer test with short suction pipe of ample area while additional inlet losses may have been introduced by the method of installation.

It is well known that a "vortex action" takes place at and surrounding the rotating impeller hub, which, increasing the temperature of the water due to friction at that point, increases also the vapor pressure and with insufficient water seal on the inlet the water vaporizes at the increased temperature and pressure at the impeller hub. This prevents the entrance of water through the zone which the vapor occupies.

Apparently in the case cited by the author there was excessive inlet loss as is evidenced by the fact that later on there

was no difficulty in delivering 1200 G.P.M. to the impeller with a water seal of only 3½ feet, at presumably the same temperature.

The introduction of a stationary radial baffle at the impeller inlet would produce the same desirable result as the guiding cone described by the author and would have the further desirable effect of reducing surface friction while it is usually less expensive construction. Similar results may be expected by the introduction of the cone or the baffle when handling cold water under extreme suction lift conditions—the effect being to eliminate the vortex and consequent reduction of inlet losses.

This writer notes that the shape of the inlet vane shown in Figure 21 is different to that shown in Figure 22. The latter illustrates the changes made in the impeller for the purpose of increasing its capacity. As that shown by Figure 22 indicates the more desirable shape for majority of conditions it would be interesting to learn whether the shape of inlet vane was changed at same time as the guide cone was introduced.

NOTES.

VESSELS UNDER CONSTRUCTION, UNITED STATES NAVY.

PROGRESS AS OF MARCH 31, 1926.

Type Number and Name	Contractor	Per Cent of Completion April 1, 1926 Total	on Ship	March 1, 1926 Total	on Ship	Contract date of Completion	Probable date of Completion
AIRPLANE CARRIERS.							
Lexington	Beth. SB. Corp. (Fore River)	77.2	75.3	75.3	73.1		4/1/27
Saratoga	New York SB. Corp.	81.8	81.2	79.8	79.2		12/1/26
AUXILIARIES.							
Holland	Puget Sound Navy Yard	93.	92.6	90.4	89.8		6/1/26
FLEET SUBMARINES.							
V-3 (SF6)	Portsmouth Navy Yard	99.5	99.2	97.1	96.3		6/1/26
V-4 (SM1)	Portsmouth Navy Yard	32.	29.8	31.3	29.3		4/1/27
RIVER GUNBOATS.							
PG43	Kiangnan Dock & Eng. Wks.	.2	.....	.....	.....	.....	.....
PG44	Kiangnan Dock & Eng. Wks.	.2	.....	.....	.....	.....	.....
PG45	Kiangnan Dock & Eng. Wks.	.2	.....	.....	.....	.....	.....
PG46	Kiangnan Dock & Eng. Wks.	.2	.....	.....	.....	.....	.....
PG47	Kiangnan Dock & Eng. Wks.	.2	.....	.....	.....	.....	.....
PG48	Kiangnan Dock & Eng. Wks.	.2	.....	.....	.....	.....	.....

Authorized but not under construction.

Pensacola (CL24)—Plans and specifications issued to New York Navy Yard for preparation of estimates.

Salt Lake City (CL25)—Bids opened April 1, 1926 but contract not yet placed.

V-5—Plans and specifications issued to Portsmouth Navy Yard for preparation of estimates.

V-6—Plans and specifications issued to Mare Island Navy Yard for preparation of estimates.

Authorized by Act of August 29, 1916, but no funds for construction appropriated.

12 Destroyers Nos. 348 to 359 inclusive.
1 Transport No. 2.
3 Fleet Submarines Nos. 169 to 171 inclusive.
1 Neff Submarine No. 108.

Authorized by Act of December 18, 1924, but no funds for construction appropriated.

6 Light Cruisers Nos. 26 to 31 inclusive.

COAST GUARD CUTTER FOR ARCTIC SERVICE.

NEW VESSEL FOR BERING SEA PATROL TO BE STRONGEST, FOR ITS SIZE, EVER BUILT—DIESEL ELECTRIC MACHINERY TO BE INSTALLED.

Of all the varied and difficult services performed by the Coast Guard cutters, there is probably none of a more arduous and trying nature and calling for the highest qualities and type of both ships and their crews than the Bering Sea patrol. For many years this service has, in large part, been performed by the cutter *Bear*. Built in Greenock, Scotland, in 1874, especially for Arctic service as a whaler, the *Bear* has far outrun the span of life usually allotted to ships. Her long and useful life speaks volumes for the thorough work of her builders and the skill and care with which she has been handled and maintained in service, but like all products of human hands, she has finally outlived her usefulness and will soon disappear from the world of living ships.

To replace the *Bear* the vessel which is the subject of this article has been designed by the Construction and Repair Department of the United States Coast Guard Service at Washington. Plans and specifications of the new cutter are now in the hands of the shipbuilders and bids for her construction are shortly to be opened.

In the building of this vessel a procedure differing from the ordinary is to be followed. The contractors are called upon to bid on the construction of the hull and the installation only of the propelling machinery, the latter having already been purchased by the government and together with a considerable amount of equipment, including such items as boats, hawsers, navigating oufit, radio outfit, china and bed linen will be delivered free of cost to the contractor at the latter shipyard.

An inspection of the plans reveals a ship of most unique design, quite different from any preceding Coast Guard vessels. The principal dimensions and particulars follow:

Length overall	216 feet 0 inches
Length on waterline	200 feet 0 inches
Beam, molded	39 feet 0 inches
Depth, molded at lowest point of sheer	24 feet 9 inches
Draft, designed mean, with 64,000 gallons fuel oil, 50 tons coal, 16,000 gallons fresh water, 60 tons stores and 20 tons cargo	15 feet 0 inches
Displacement at above draft	2050 tons
Speed, designed trial, with 30,000 gallons fuel oil, 25 tons coal, 8000 gallons fresh water, 30 tons stores and 10 tons cargo	11 knots per hour
Shaft horsepower, estimated, at above speed and loading	1000

NOTE—The figures given above under "draft" represent the approximate total capacities of bunkers and tanks.

So far as the hull is concerned the outstanding requirement governing the design is ability to navigate in heavy ice and withstand the enormous pressures of entangling ice flows. To this end the forefoot is cut away to above the waterline and all the shell plating at or near the latter is of exceptionally heavy scantling, with a maximum thickness of 1¼ inches at the bow. Supporting this heavy shell is a framing system of extraordinary strength and stiffness. The transverse frames, spaced 24 inches, are *all* of the built up or web type, up to the main deck, and are 16 inches in depth. To resist the heavy transverse stresses due to ice pressure, the upper, main and berth decks are each completely plated and have heavy channel beams on every frame, the berth deck beams, which will be called upon to withstand the heaviest pressure, being 10 inches in depth.

To the shipbuilder, the midship section is of particular interest as indicating the detail construction of what is, perhaps, for its size, the strongest and heaviest steel hull which has ever been projected.

The upper deck, outside of the houses, is to be planked with 3-inch yellow pine or douglas fir as is also the boat deck, and 2½-inch decking will be laid on the bridge deck. Decks protected from the weather will, in general, be covered with Armstrong battleship linoleum. Decks in toilets and washrooms will be covered with asphalt or bitumastic composition and in the galley with corrugated tile.

The bulkheads in quarters will be, in general, of wood. Insulation consisting of compressed cork, covered with linoleum or bolted to galvanized steel sheathing is to be fitted as necessary to the ship's sides, and to casings and bulkheads in way of living quarters. Wood ceiling, paneled in cabin and wardroom and beveled, tongued and grooved in wheel house, chart room and similar spaces will also be fitted.

The deck and navigating outfit is necessarily very complete. The following boats, all of the Coast Guard standard type, will be carried:

One 26-foot motor launch.
One 20-foot motor dinghy.
One 26-foot sailing launch.
One 27-foot whaleboat.
One 26-foot Monomoy surf boat.
One 25-foot 6-inch self-bailing surf boat.
One 18-foot dinghy.

For handling the boats, four sets of mechanical davits will be fitted.

The ground tackle will consist of three 4000-pound, one 1000-pound and one 500-pound stock anchors, the weights given being exclusive of stock; and 240 fathoms of 1¾-inch cast steel, stud link chain cables.

A Sperry commercial type gyro compass system will be installed consisting of a master compass and 6 repeater compasses. The usual magnetic compasses will also be carried as well as a Lietz electric driven sounding machine and a Submarine Signal Corporation fathometer. A Lux carbon-dioxide fire extinguishing system, furnished by Walter Kidde and Company, will be installed in the machinery spaces.

PROPELLING MACHINERY.

The propelling machinery of the new cutter marks an even greater departure from previous Coast Guard practice than the hull. The Diesel-electric system of propulsion is to be used and all auxiliaries and deck machinery will be electrically driven, steam being used only for heating, cooking and as a visual signal in conjunction with air whistles or horns. Even the feed pump for the two small steam boilers will be electrically operated.

The main propelling unit will consist of two 6-cylinder, 4-cycle, single acting McIntosh and Seymour Diesel engines, each developing 600 brake horsepower at 200 revolutions per minute and directly connected to a General Electric direct current generator, having an output of 410 kilowatts at 250 volts. Directly attached to extensions of the generator shafts will be two direct current exciters, each having an output of 75 kilowatts at 120 volts.

The two generators will drive a single 500-volt, double armature motor having an output of 1000 shaft horsepower, and directly connected to the single propeller shaft by a magnetic clutch, the latter being located between the motor and the main thrust bearing.

The line and propeller shafts, 10 inches and 11 inches in diameter, respectively, will drive a four-bladed, built up bronze propeller 10 feet in diameter.

AUXILIARIES.

As already remarked, all auxiliary machinery, below and on deck, will be electrically operated by current from the main or auxiliary generators. Of the latter, which are all of General Electric manufacture, two have an output of 60 kilowatts each at 120 volts and 500 revolutions per minute and are driven by 6-cylinder, 4-cycle Winton Diesel engines of 90 brake horsepower.

The principal function of these large auxiliary generators will be to furnish excitation for the two main generators and the propelling motor in case of emergency. Each is of sufficient capacity for all of this work. In addition to the two principal auxiliary generating sets, a third, consisting of a Hill 25 brake horsepower Diesel mounted on a common base with a 120-volt, 18-kilowatt generator will be installed for furnishing power when the larger generators are not in use.

In order to utilize "shore" alternating current when in port, an 18-kilowatt General Electric motor-generator will be installed.

The following auxiliaries will also be installed:

3 Warren centrifugal engine cooling, water circulating pumps, 250 gallons per minute capacity.

2 Warren centrifugal fire pumps, 250 gallons per minute capacity.

1 Warren centrifugal fuel oil heating, water circulating pump, 35 gallons per minute capacity.

1 Warren centrifugal brine circulating pump, 15 gallons per minute capacity.

1 Warren centrifugal fresh water pump, 25 gallons per minute capacity.

1 Quimby screw fuel oil transfer pump, 45 gallons per minute capacity.

2 Quimby screw boiler feed pumps, 7 gallons per minute capacity, each.

4 Trimount rotary hand pumps for oil and water transfer, 20 gallons per minute capacity, each.

1 Goulds plunger hand boiler feed pump, 8 gallons per minute capacity.

1 De Laval centrifugal fuel oil purifier, 250 gallons per hour capacity.

1 De Laval centrifugal lubricating oil purifier, 300 gallons per hour capacity.

7 Schutte & Koerting water jet bilge eductors, 20,000 gallons per hour capacity, each.

7 Schutte & Koerting water jet bilge eductors, 2000 gallons per hour capacity, each.

1 Schutte & Koerting water jet portable water eductor, 5000 gallons per hour capacity.

1 Schutte & Koerting boiler injector, 200 gallons per hour capacity.

2 Coatesville vertical firetube heating boilers, 25 horsepower, 100 pounds per square inch working pressure.

1 York Manufacturing Company ammonia refrigerating machine, 2 tons.
1 F. E. Meyers & Bro. Company hydro-pneumatic fresh water outfit, 500 gallons per hour capacity.
1 American Strombos Company compressed air signal horn.
1 motor-driven engine lathe, 16-inch swing, 8-foot bed.
1 motor-driven drill press, 20 inches.
1 motor-driven pedestal grinder.
1 motor-driven anchor windlass for 1¾-inch cable.
1 motor-driven gypsy, 20-inch warping head.

All of the motors for operating auxiliaries will be of General Electric manufacture.

The foregoing notes and abstracts from the specification which have just been issued from Washington will serve to give a fairly clear idea of the ship which it is proposed to construct. Although, as has been remarked before, the method to be followed in building this vessel differs considerably from former practice, in that the government is itself to furnish the propelling machinery and in consequence must, in large part, assume the responsibility for satisfactory performance, the specifications are clear and explicit, and there appears no reason to doubt that the final results will be satisfactory to all concerned.—"Marine Engineering and Shipping Age," March, 1926.

The Newport News Shipbuilding and Dry Dock Company, of Newport News, Va., has been awarded the contract for construction of the hull of the new Coast Guard cutter *Bear*, at a cost of $585,000. The vessel will replace the famous *Bear*, which for years was noted for its rescue work in northern Pacific waters.

ELECTRIC PROPULSION OF SHIPS.*

By Eskil Berg, General Electric Company.

(This paper deals largely with the application of the turbo-electric drive to a high-powered passenger vessel and gives steam consumptions, electrical losses and weights for an equipment of 17,000 S.H.P. which is actually under construction.

At the commencement of the Paper the Author deals generally with the electric drive, comparing it with a geared drive, after which he proceeds to discuss the particular example and describes the equipment and the control gear. The Author then gives a list of advantages which he claims for this form of drive, and in conclusion he briefly mentions the application of electric transmission for propelling purposes with a Diesel engine as a prime mover.)

Electric propulsion can no longer be considered in the experimental stage. Its use by the United States Navy during the last ten years, and its application to a few, rather large cargo steamers, as well as several other types of ships, have proved this method successful, reliable, economical and as far as the upkeep of the electrical apparatus is concerned it has been practically nil.

For large powers where steam is considered, the steam turbine must be used on account of its high efficiency as compared with any other steam prime mover.

Steam turbines, in order to obtain the best efficiency, must be designed for relatively high speed (R.P.M.), which makes it necessary to have either a mechanical speed reduction or an electrical transmission from the high speed turbine shaft to the slow speed propeller shaft.

* A paper presented to the North East Coast Institution of Engineers and Shipbuilders, New Castle, England, February 12, 1926.

SHIPS WITH ELECTRICAL PROPULSION.

Name.	Type of Ship.	Year in Service.	Type of Drive.	Speed.	S.H.P.	Propeller.		Turbine.		Generators.		Pressures.		Superheat.
						No.	R.P.M.	Stages	R.P.M.	No.	Voltage.	Gauge	Vac.	Deg. F.
S.S. Joseph Medill	Fire boat	1908	Turbine-electric	11·7 miles	500	2	179	2	1,800	2	275	160	27	0
S.S. Graeme Stewart	Fire Boat	1908	Turbine-electric	11·7 miles	500	2	179	2	1,800	2	275	160	27	0
U.S.S. Langley (*ex*-Jupiter)	Airplane Carrier	1913	Turbine-electric	14·5 knots	5,400	2	110	9	1,900–2,100	1	2,300–2,420	175	27·5	0
U.S.S. New Mexico	Battleship	1918	Turbine-electric	21 knots	28,000	4	161	10	2,130	2	3,000–4,242	250	28·5	50
U.S.S. California	Battleship	1921	Turbine-electric	21 knots	28,000	4	170	10	2,065	2	3,000–4,242	250	28·5	0
U.S.S. Maryland	Battleship	1921	Turbine-electric	21 knots	28,000	4	170	10	2,065	2	3,000–4,242	250	28·5	0
U.S.S. West Virginia	Battleship	1921	Turbine-electric	21 knots	28,000	4	170	10	2,065	2	3,000–4,242	250	28·5	0
S.S. Eclipse	Cargo	1920	Turbine-electric	11 knots	3,000	1	100	8	3,000	1	2,300	200	28·5	200
S.S. Invincible	Cargo	1921	Turbine-electric	11 knots	3,000	1	100	8	3,000	1	2,300	200	28·5	200
S.S. Archer	Cargo	1921	Turbine-electric	11 knots	3,000	1	100	8	3,000	1	2,300	200	28·5	200
S.S. Independence	Cargo	1921	Turbine-electric	11 knots	3,000	1	100	8	3,000	1	2,300	200	28·5	200
S.S. Victorious	Cargo	1921	Turbine-electric	11 knots	3,000	1	100	8	3,000	1	2,300	200	28·5	200
S.S. Cuba	Pass.-Cargo	1920	Turbine-electric	17 knots	3,000	1	100	8	3,000	1	1,150	175	27·5	150–250
M.S. Mariner	Trawler	1920	Diesel-electric	10 knots	400	1	160–200	—	—	2	250	—	—	—
S.S. Tampa	Coast Guard Cutter	1921	Turbine-electric	16 knots	2,600	1	130	8	3,000	1	2,300	200	28	75
S.S. Haida	Coast Guard Cutter	1921	Turbine-electric	16 knots	2,600	1	130	8	3,000	1	2,300	200	28	75
S.S. Mojave	Coast Guard Cutter	1921	Turbine-electric	16 knots	2,600	1	130	8	3,000	1	2,300	200	28	75
S.S. Modoc	Coast Guard Cutter	1921	Turbine-electric	16 knots	2,600	1	130	8	3,000	1	2,300	200	28	75
S.S. San Benito*	Pass.-Cargo	1921	Turbine-electric	12·5 knots	3,000	1	100	9	3,000	1	1,100	190	27·5	200
H.I.J.M.S. Kamoi	Fuel Ship	1922	Turbine-electric	15 knots	8,000	2	120	10	2,400	1	2,300	250	28·5	150
M.S. Fordonian	Cargo (Great Lakes)	1922	Diesel-electric	9 knots	850	1	120	—	—	2	500	—	—	—
M.S. Golden Gate	Ferryboat	1922	Diesel-electric	12 miles	750	2	145–180	—	—	2	500	—	—	—
M.S. Golden West	Ferryboat	1923	Diesel-electric	12 miles	750	2	145–180	—	—	2	500	—	—	—
S.S. Hayward	Ferryboat	1923	Turbine-electric	17 miles	1,200	2	100–125	3	3,600	1	500	210	28·5	50
S.S. San Leandro	Ferryboat	1923	Turbine-electric	17 miles	1,200	2	100–125	3	3,600	1	500	210	28·5	50
M.S. Standard Service	Tanker	1923	Diesel-electric	9 knots	600	1	130	—	—	2	460	—	—	—
M.S. Alaskan Standard	Tanker	1923	Diesel-electric	9 knots	600	1	130	—	—	2	460	—	—	—
S.S. W. R. Hearst	Ferryboat	1923	Turbine-electric	16 miles	2,200	2	123–176	8	3,240	1	2,300	250	28·5	200
S.S. Rodman Wanamaker	Ferryboat	1923	Turbine-electric	16 miles	2,200	2	123–176	8	3,240	1	2,300	250	28·5	200
S.S. Geo. W. Loft	Ferryboat	1924	Turbine-electric	16 miles	2,200	2	123–176	8	3,240	1	2,300	250	28·5	200
M.S. Twin Ports	Cargo (Great Lakes)	1923	Diesel-electric	10 miles	500	2	180	—	—	2	230	—	—	—
M.S. Twin Cities	Cargo (Great Lakes)	1923	Diesel-electric	10 miles	500	2	180	—	—	2	230	—	—	—
U.S.S. Lexington	Airplane Carrier	†	Turbine-electric	33 knots ‡	180,000	4	317	13	1,800	4	5,000	265	28·5	50
U.S.S. Saratoga	Airplane Carrier	†	Turbine-electric	33 knots	180,000	4	317	13	1,800	4	5,000	265	28·5	50
M.S. La Playa*	Pass.-Cargo	1924	Diesel-electric	12·5 knots	2,500	1	95	†	—	4	600	—	—	—
M.S. La Maria*	Pass.-Cargo	1924	Diesel-electric	12·5 knots	2,500	1	95	†	—	4	600	—	—	—
M.S. Hawaiian Standard	Tanker	1925	Diesel-electric	9·7 knots	600	1	130	—	—	2	230	—	—	—
M.S. Van Dyke, No. 1	Tug and Fire Boat	1924	Diesel-electric	10 knots	370	1	120	—	—	2	125	—	—	—
M.S. Van Dyke, No. 2	Tug and Fire Boat	1924	Diesel-electric	10 knots	370	1	120	—	—	2	125	—	—	—
M.S. Van Dyke, No. 3	Tug and Fire Boat	1924	Diesel-electric	10 knots	370	1	120	—	—	2	125	—	—	—
M.S. Brilliant	Tanker	1924	Diesel-electric	10 knots	370	1	120	—	—	2	125	—	—	—
M.S. E. M. Coots	Ferry	1925	Diesel-electric	8 miles	360	2	Sidewheels	—	—	2	230	—	—	—
S.S. T. W. Robinson	Bulk Freighter	1925	Turbine-electric	13·8 miles	3,000	1	100	8	3,450	1	1,150	300	28·5	200
M.S. Lake Weir	Dredge	†	Diesel-electric	10 miles ‡	1,000	2	130	—	—	2	240	—	—	—
M.S. U.S. Engineer Corps	Towboat	†	Diesel-electric	—	100	1	Sternwheel	—	—	1	230	—	—	—
M.S. U.S. Engineer Corps	Towboat	†	Diesel-electric	—	100	1	Sternwheel	—	—	1	230	—	—	—
M.S. U.S. Engineer Corps	Towboat	†	Diesel-electric	—	225	1	Sternwheel	—	—	2	230	—	—	—
M.S. Bear	Coast Guard Cutter	†	Diesel-electric	12 knots‡	1,000	1	120	—	—	2	500	—	—	—
S.S. International Mercantile Marine Co.	Pass.-Cargo	†	Turbine-electric	18 knots	17,000	2	120	16	2,888	2	4,000	550	28·5	100

* Vessels equipped by the British Thomson-Houston Company, Limited, of Rugby, England.

† Under construction.

‡ Designed speed.

Mechanical gears for large powers require a multitude of turbines and pinions, complicated steam piping, and cross-over connections between the various turbines, as well as a multitude of turbine gland packings.

Reversing turbines are a source of great loss, not only friction loss, but, as they necessarily make the turbine longer (which makes the distance between bearings longer), a larger diameter shaft for the same deflection is involved, thus increasing the packing losses as well as bearing losses.

With mechanical gears, the turbines adopted have generally to be of slower speed (R.P.M.) than that which would give the most economical design for the horsepower required. This last item is important as, in almost every case, the slower speed turbine with the features mentioned above will involve a loss in turbine economy that is a great deal more than the total electrical transmission loss and, in some cases, it may be more than double the electrical losses.

Steam turbines driving generators are constantly being tested by manufacturers as well as customers, and their performance is a matter of definite knowledge.

In electric ships, the results obtained have always checked with the test results of the turbo-generators and motors as made at the factory. Full advantage can always be taken of any new improvement in turbine design which develops in the enormous field of turbo-generators for central station use. In other words, the very latest turbine designs can always be used.

The enormous advantage in economy by the use of high steam pressure, steam temperature, steam extraction for heating feed water, etc., can be secured in the electric ship without any complication and with practically no special divergence from standard modern steam station practice.

Many stations are now in successful operation using 550-pound boiler pressure and steam temperature of 725 degrees which produce one kilowatt hour at the switchboard with about one pound of coal of 14,500 B.T.U. This is not a record of a few hours' operation but is a weekly record and includes the whole power plant, taking into consideration load factor, cleaning boilers, etc.

What this means on board ship may best be illustrated by the fact that a large luxurious liner using 50,000 H.P. can be operated with a total fuel consumption of less than 300 tons of fuel oil per day, including everything on board the ship. (See paper by author read before the A. S. M. E., April 1, 1924.)

Electric propulsion of large warships has proved its great success and economy and needs no further discussion in this paper.

A field which has not yet been covered, and to which electric propulsion is particularly well suited, is the propulsion of large liners.

Just recently the International Mercantile Marine has closed a contract with the Newport News Shipbuilding Company for an electrically driven, twin-screw liner, having 17,000 H.P. The propelling machinery is to be built by the General Electric Company of the United States of America. A complete analysis of the electric propelling machinery in this ship will be given.

The dimensions and particulars of the ships are as follows:

Length, 600 feet; Beam, 80 feet; Draft, 30 feet.
Displacement, 22,000 tons.
Propeller speed, 120 R.P.M.
Horsepower, 17,000.
Speed, 18 knots.
Propellers, 18 feet 6 inches diameter; 17 feet 3 inches pitch.

Steam Conditions:—Steam pressure at throttle, 250 pounds; superheat at throttle, 100 degrees; vacuum, 28½ inches.

Boilers:—12 Babcock & Wilcox oil burning marine boilers having each 4600 square feet of heating surface and 1344 square feet of superheating surface.

Electrical Equipment:—

2 Turbine driven, alternating current generators having a rating of ATB 2—6600 K.W.—2888 R.P.M.—4000 volts.
2 Main synchronous induction motors for direct connection to the propeller shafts, having a rating ATI 48—6600 K.W.—120 R.P.M.—4000 volts.
4 500 K.W. direct current generating sets.
1 Control board.
Cables.

Weights:—In this case no attempt has been made to reduce the weights of the apparatus from that used for land purposes.

	Pounds
The weight of the two turbo-generators, including all accessories, tools, etc., is	306,000
The 2 main propelling motors weigh	338,000
Complete control equipment, including cables	17,000
	661,000

Weight per shaft horsepower, 39 pounds.

In this connection it may be of interest to note that electric propelling machinery has also been laid out for a light cruiser having 120,000 H.P. In this case weight was of the greatest importance, and the equipment weighed 18 pounds per shaft horsepower.

Price:—The price of the electrical propelling equipment completed in this case is substantially the same as the geared turbine equipment proposed for this ship, but on account of simpler and less costly foundations, piping, oiling systems, etc., it was found to be cheaper than any other method of propulsion proposed.

Control apparatus consists of necessary contactors for closing the electrical circuits between the generators and motors. It will be operated by means of 4 levers (2 for each motor). In addition, there will be 2 small levers for regulating the speed of the main turbines.

The 2 main levers for each propulsion motor consist of:

One for regulating the excitation of the generator and the motor;

The other is the reversing lever, which has three positions, "ahead," "stop," "astern."

The field lever has four definite positions, "off," "1," "2," "Run."

On position "1" double excitation is applied to the generator alone. In this position, with the reverse lever on either "ahead" or "astern," the propelling motor is started or reversed as a pure squirrel cage induction motor.

In position "2" normal excitation is applied to the rotor field, leaving double excitation on the generator. In this position the motor phases in and becomes a synchronous motor.

Position "Run" reduces the excitation of the generator to normal.

Speed variation of the propeller revolutions in either direction is obtained by varying the turbine speed, and is controlled by the operator from the small speed lever on the control board.

One turbo-generator operation:—Normally both turbo-generators are in operation. Should, however, it be desired at any time to operate the ship at three-fourths speed or lower for a considerable length of time, only one generating unit need be used and both propelling motors can be operated

from the control board exactly in the same way as when both units are running. This feature should prove most economical should the ship ever be used for cruises during the slack season as in this case almost 15 knots can be obtained.

Reversing Ship at full speed:—

1st:—Set levers controlling speed governor on turbine to low speed position.

2nd:—Put both field levers to "off" position.

3rd:—Pull the reverse levers from the "ahead" position to the "astern" position.

4th:—Pull the field levers to the "1" position and hold them there until the propeller speed indicators show that the propellers have reversed and come up to constant speed in the opposite direction.

5th:—Pull the field lever to notch 2 and note by instrument when motors pull into step.

6th:—Then put field levers to "Run" position.

7th:—Move turbine speed lever to any desired propeller R.P.M.

Reversing Torque:—The motors and the generators are designed to give the necessary torque to reverse the propellers from 120 R.P.M. ahead to 120 astern in 20 seconds. Quicker reversal can be had but is not considered advisable.

Steam Consumption:—When operating with 250 pound gauge pressure and 100 degrees superheat at the throttle and with 28½ inches of vacuum, the steam consumption per shaft horsepower delivered to the propeller shaft (including all losses in generators; motors, cables, etc.) will be 9.5 pounds when delivering 17,000 H.P. to the propeller shaft at 120 R.P.M.

When delivering 13,500 H.P. at 110 R.P.M. under the same steam conditions the consumption will be 9.9 pounds. Considerable margin is included in these figures to take care of sea conditions and inaccuracy of tests.

Fuel Consumption:—If electrically driven auxiliaries are used, the fuel consumption per shaft horsepower hour, including everything on the ship, figures at 17,000 H.P. to be 0.85 pound per shaft horsepower and, at 13,500 H.P. 0.90 pound. In these calculations slightly over 1000 K.W. direct current load has been assumed at 17,000 as well as 13,500 H.P. conditions.

Superheat:—This ship will be operated with only 100 degrees superheat. The design of the turbine is, however, made to use efficiently and safely 300 degrees of superheat should it at a later date be desired to increase the superheat or if new additional ships should be built.

The fuel saving by going to 200 degrees over 100 degrees is 5 per cent and 300 degrees over 100 degrees is 9 per cent.

Electric propulsion is particularly well adapted to high steam temperatures, as all the reversing is done electrically and no undue heat stresses or temperature changes take place in the turbine itself—as is the case when mechanical gears are used.

Efficiency of Transmission with Electric Drive:—In making comparison of transmission efficiency between geared turbine drive and electric transmission we must not compare the gear efficiency alone with the efficiency of the electric transmission. Such comparison is very misleading. In the case of this liner, for instance, if gears were to be used, the turbine design, as well as the gear design, would have to be compromised in order to compensate for the limitations imposed by the gears.

Assume single reduction gears which were, as a matter of fact, first proposed for this vessel. It was found necessary to build the turbine in three casings in order to get proper pinion design. The R.P.M. of the turbines proposed was less than half that proposed with electric drive.

Steam seals and the number of packings were three times as many as for electric drive. With electric drive it is possible to design a much more efficient turbine because we have:—

1. Increased efficiency due to the use of higher speed turbine.
2. Increased efficiency due to the elimination of the reversing turbine losses, as the same do not exist with electric drive.
3. Increased efficiency due to the elimination of all crossover connections between turbines, as the electric turbine is built in one casing—making an ideal turbine design.
4. Increased efficiency due to the lesser number of packing glands and steam seals.
5. Increased efficiency due to closer running clearances in turbine for the same factor of safety, as no excessive temperature changes take place while reversing or maneuvering.

From the above it will be seen that the actual transmission efficiency is not an index as to the comparative economy of the two methods.

Electrical Losses:—At 17,000 H.P. load and 120 R.P.M. of the propellers, the generator has an efficiency of 96.75 per cent excluding the excitation which in this case will be 32 kilowatt per generator or less than ½ per cent.

The motors will have an efficiency of 97.5 per cent excluding excitation which amounts to 45 K.W. per motor, or 0.68 per cent.

The loss in cables and control amounts to 0.04 per cent.

The total transmission loss is, therefore, excluding excitation, about 5.7 per cent, or, if excitation is included, about 6.75 per cent.

For larger powered ships the efficiency becomes, of course, better and a total loss of slightly less than 6 per cent may be expected.

At reduced speed of the vessel the efficiency of the generator and motors remains practically constant, as the voltage is then reduced in about the proportion of the square root of the horsepower so that at three-quarters speed the voltage is reduced from 4000 volts to about 3000 volts, and the transmission loss is only increased from 6.75 per cent to 7.4 per cent although the load is reduced from 17,000 H.P. to about 7000 H.P. The loss in gear efficiency, under similar conditions, would be about 2 per cent.

Advantages of Electric Drive for a Large Liner:—

1. Electric transmission gives a simple and practical means of speed reduction between the high speed turbine and the slow speed propeller in almost any ratio which may be desired.
2. Reversal. It affords means of reversal by simple change of electrical connections, without changing the direction of rotation of the turbine.
3. Reversing torque. Any desired reversing torque up to the full power of the turbine can be obtained without affecting the efficiency of the equipment in the forward direction.
4. Reversal is obtained with a minimum amount of steam drawn from the boilers, as the full inertia of the turbo-generator is first used in the act of reversal and, in fact, in most cases reversal is obtained without drawing any steam from the boilers, and steam is only used from the boilers to accelerate the propeller speed in reverse rotation.
5. Noise. Electrically driven vessels can be made noiseless, whereas gears always make more or less noise.
6. Power Measurements. Electric drive makes it possible to obtain accurate data, either by instantaneous readings or by recording instruments, of the load on the propeller shaft under various conditions of sea and wind. It eliminates the use of tortion meters. The total horsepower hours are recorded during any given time or journey. It makes it possible to have a real check between fuel consumption and horsepower hours developed—which means a constant and continuous check on economy.

7. Maintenance. Experience in this respect on the electrically built ships by the General Electric Company and the British Thomson-Houston Company has shown that the upkeep of the electrical machinery has been practically nil so far.

8. Economical Cruising Speeds. The tendency lately is to put some of the large and medium size liners on winter cruises or other cruises when the season is slack. With electric drive about three-quarters speed can be obtained by the use of only one of the two power units in operation, which means a very large saving in fuel and attention. The steam consumption per shaft horsepower will be practically the same as that of maximum speed.

9. Repairs. In the design of the generators and motors special precaution has been taken so that, should for any reason a damage occur to any part of the windings, this can quickly be repaired, or temporary quick repairs can be made, so that the ship can proceed at slightly reduced speed. Multiple circuits are used in the generators and motors so that in many cases (for instance, if a coil is damaged) it can simply be cut in two by a hack saw, the ends insulated and the loss in power will be only the ratio of this one circuit to the total number of circuits in multiple. So, for instance, if the motor has 10 circuits in multiple, and one circuit is cut out, the power is reduced only one-tenth, which will be practically unnoticed, and the repair should take only a few minutes. If several coils are damaged, a jumper can be connected across the damaged coils, and the vessel can proceed. Electrical repairs are always easily made when fully understood.

10. Location of Apparatus. Electrical drive in many cases may afford important advantage because the generating unit, or units, can be placed in any convenient place near the boilers. They can be mounted at any height desired. The condensers can be put directly under the turbines, thus reducing the steam and the exhaust piping to a minimum. Chances of air leaks, which are so detrimental to obtaining good vacuum, are reduced to a minimum. The propelling motors can, in many instances, be put near to the propellers, eliminating long, expensive shafting and the bearing losses incident to the same. Shaft alleys can be eliminated, thus saving hold space and making cargo handling much easier and convenient.

11. Design of Turbine. The turbine for electric transmission can be built more efficient. No reversing wheels are required, thus the turbine becomes shorter, which gives smaller shaft, less diaphragm packing leakage, packing and bearing losses. Since there is no heat distortion during reversal, smaller clearances can be used with the same factor of safety, all of which goes toward greater economy.

12. High Steam Pressure and Superheat. With electric drive the adoption of the above is ideal, and full advantage can be taken of the economy derived by its adoption. As the turbine is never reversed, it cannot have extreme temperature changes.

Diesel Electric Drive:—Since the adoption of Diesel electric propulsion for the beam trawler *Mariner*, which was put in operation early in 1920, there has been a marked progress in the application of this type of propulsion over a wide range of vessels—there now being some forty-five ships using this type of drive, in operation, or under construction. These include tug boats, ferry boats, cargo boats, oil tankers, dredgers and Coast Guard cutters. The largest Diesel electric drive equipment installed at present in any single ship is 2500 shaft horsepower. Several operators have considered Diesel electric propulsion in considerably larger sizes, and it would not be surprising if in the near future some one would go ahead with this form of propulsion in larger capacities.

The most promising field at present, however, seems to be in the smaller craft, especially where a large percentage of installed horspower is required for driving auxiliaries. This permits the same prime mover to furnish

the power for both auxiliaries and propulsion. Examples of this are tankers where electrically driven pumps are used for loading and unloading cargo. The engine driven generators furnish the power for driving the ship and the same generators are used for furnishing power for driving the cargo pumps in loading and unloading Diesel electric has also proven itself suitable for dredgers.

Tug boats offer another example where Diesel electric drive is particularly effective, and tests recently conducted by the Atlantic Refining Company have indicated beyond a doubt that the Diesel electric drive is far superior in tug boat service to either steam or Diesel direct drive.

There have been a number of articles written both in favor and against Diesel electric drive and, without going into the advantages or disadvantages, it is admitted that this form of propulsion has its limitations, the same as other forms of propulsion, and is much more suited for some types of ships than others. Operators who have Diesel electric driven ships seem very well pleased with results obtained, and most of them, when ordering additional tonnage, are installing Diesel electric drive, which is after all the strongest argument for the success of this form of propulsion where it has been applied.

It will be seen from the foregoing list that great headway has been made in the application of electricity for the propulsion of ships. With the advent of the adoption of high steam pressure, superheat, steam extraction for feed heating, as well as possible reheating between stages, electric propulsion makes the simplest and most ideal installation, and results can be obtained equal to, or better than the best Diesel when total fuel consumption of the whole ship is taken into consideration.

Electric propulsion is also ideal for the mercury turbine should this be considered for ship propulsion. Results so far obtained on land prove conclusively that this type of prime mover is more economical than the best of Diesel engines produced so far.

THE R. M. S. P. CO.'S MOTOR LINER *ASTURIAS*.

The *Asturias* is by far the largest vessel of the R. M. S. P. Co.'s fleet, and it is understood that when the sister ship *Alcantara* is completed, it will enable a 10 days' service to be maintained between Southampton and Buenos Aires. It is impossible to mistake the *Asturias* for any other ship, her low funnels and cruiser stern giving a distinctive and undoubtedly attractive appearance. These funnels measure about 35 feet by 25 feet, but only one is used for the reception of all the engine silencers and for the uptakes from the heating boilers.

The following are the leading particulars of the new vessel:

Gross tonnage	22,137
Length between perpendiculars, feet	630
Length overall, feet and inches	655—8
Moulded beam, feet	78
Depth, feet	45
Complement of passengers and crew	1,800
Number of engines	2
Number of cylinders of each engine	8
Cylinder bore, inches	33
Piston stroke, inches	59
Normal speed, R.P.M.	115
I.H.P. each engine	10,000
B.H.P. each engine	7,500
Speed of ship	About 16½ to 17 knots

THE TRIALS.

Before proceeding with a description of the vessel and the machinery, some details of the trials and the performance of the ship on her trip from Belfast to Southampton may be given. The preliminary trials lasted in all about five days and were carried out in January, the liner being headed direct for the Clyde immediately after leaving Belfast. These tests included a 48 hours' run at full service speed, followed by progressive trials on the Skelmorlie mile, the engine revolutions varying from 92 to 122 per minute. The latter represented a very considerable overload, since an output of 23,300 I.H.P. was maintained for 1½ hours, whilst a maximum speed of 18 knots was reached.

At service speed, with the engines presumably developing slightly under 20,000 I.H.P. and corresponding to between 16½ knots and 17 knots, the fuel consumption for all purposes amounted to 70.07 tons per day, this being made up roughly of 58 tons for the main engines, three tons for the boilers, and the remainder for the Diesel-driven generators. The total engine revolutions during the preliminary trials were 700,000, at an average of 112 R.P.M., equivalent to a voyage of 2000 miles.

The propelling engines have cylinders with the same bore and stroke as those on the motor liner *Gripsholm* and those which will be fitted to the sister ship of the *Asturias* (the *Alcantara*) and the Union Castle liner *Carnarvon Castle*, whilst the two Cosulich passenger vessels of 23,500 tons gross, the *Saturnia* and *Urania*, will have machinery of equal cylinder size. It is thus not too much to say that a certain degree of standardization has already been attained in the manufacture of these big double-acting units.

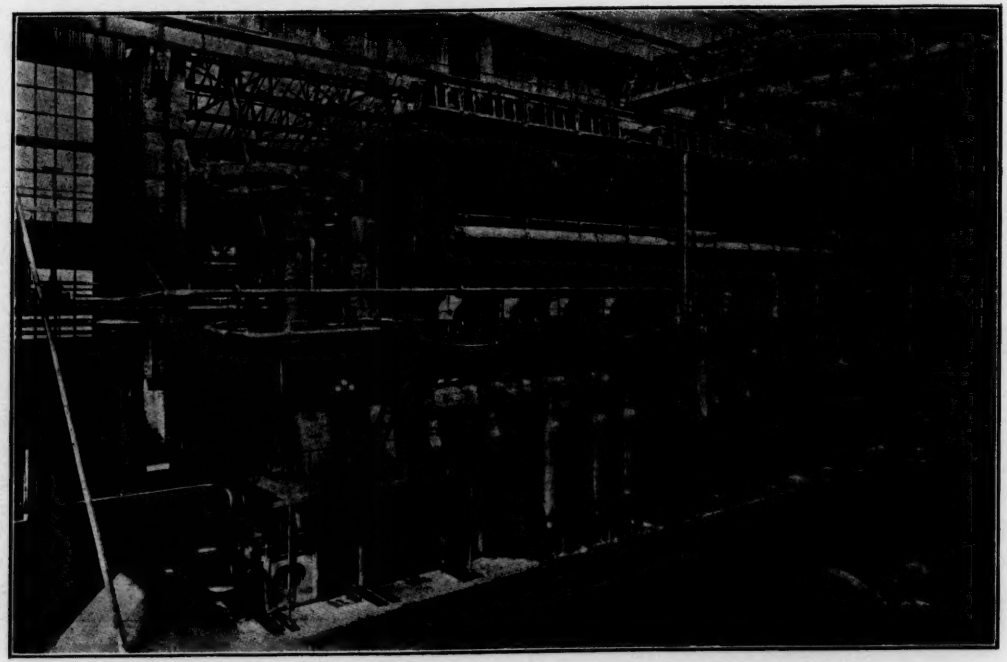
It will be remembered that in the *Gripsholm*, with six-cylinder engines, the air compressors were separately driven by Diesel engines. In the *Asturias* there are two eight-cylinder units, and each drives its own twin injection-air compressor at the forward end of the crankshaft. The main engine cylinders have a bore of 840 millimeter (33 inches) and a piston stroke of 1500 millimeter (59 inches), and each motor develops 10,000 I.H.P. The mechanical efficiency is almost exactly 75 per cent, giving a brake horsepower of 7500, and the mean effective pressure is about 70 pounds per square inch, referred to brake power.

ENGINE CONSTRUCTION.

The general construction of this type of Harland-B. and W. four-stroke double-acting engine is now fairly well known. So far as the upper covers are concerned, there is practically no difference from the single-acting design, but the lower combustion chamber is attached to one side of the bottom cylinder cover, with the exhaust valve vertically arranged at the bottom; the air inlet valve is at the top, the fuel and starting-air valves being nearly horizontal. Over the piston rod is a cast-iron sleeve with a space between the rod and the sleeve, and this sleeve passes through a gland in the bottom cylinder cover.

Tension stresses are taken by through bolts extending from the top of the cylinder covers to the bottom of the bedplate. These tie rods are in two parts with the joints made by nuts at a level just above the bottom cylinder cover. The cylinder covers are bolted together, whilst at the center, between the two groups of four cylinders, rigidity is ensured by distance bolts. It may be added that the two engines are tied together by horizontal tie rods right at the top, at each end and at two points in the center; that is to say, each group of four cylinders is held by two rods.

The horizontal camshaft actuating all the valves, both at the bottom and top of the cylinders, is chain driven from the crankshaft at the center of the engine. The level of the camshaft is at about the center of the



ONE OF THE 10,000 I.H.P. MAIN ENGINES OF THE "ASTURIAS."

cylinders. For each cylinder there are four ahead and four astern cams for the top valves, and the same number for the bottom, making 16 cams per cylinder. The operation of the upper valves is similar to that of the single-acting engine with long push rods and rollers at the bottom, to which are fixed short horizontal levers pivoted on the upper maneuvering shaft. For the lower valves the principle of the arrangement is similar, although the lever system has to be modified, owing to the position and angularity of the respective valves. There is thus a maneuvering shaft for the upper valves and one for the lower, by the rotation of which the valve-rod rollers are taken out of contact from the cams, so that the camshaft may be moved fore and aft to bring the astern cams into action.

The main controls embody merely a small reversing lever and two combined starting and speed-control levers, identical with those utilized in single-acting engines.

On operating the reversing lever, the servo motor is brought into action, rotating the maneuvering shafts and thus freeing the camshaft for moving fore and aft; after this has been effected, a further rotation of the maneuvering shafts brings the cams again into contact with the lever rollers. The two starting levers are then pushed to the starting position, and air is admitted to the top or bottom of the cylinders. One lever is next moved to the running position, allowing starting air to be shut off and the fuel pumps to come into action and supply oil through the fuel valves at the top of the cylinders. The second lever is then similarly pushed forward and the engine is operating on fuel, all starting air having been shut off. By the same two levers the speed control is effected by acting upon the suction valves of the fuel pumps in the usual manner, these fuel pumps being located above the levers, as seen in the illustration.

For the cylinders and jackets, fresh-water cooling is employed, and for the pistons, lubricating oil, which passes up through a telescopic pipe to a reservoir, thence down to a branch piece on the crosshead. It rises up to the cooling space in the upper portion of the piston, through the annular space between the piston rod and the sleeve previously mentioned. The oil is returned through an axial hole in the center of the piston rod to a thermometer pocket within the crankcase, and thence to the sump. It passes through a strainer into a drain tank, is lifted by a forced-lubrication pump and delivered to a cooler. The oil used for lubricating is the same as that employed for cooling, but the two circuits are in parallel.

There has been some discussion lately concerning the question of oil fumes in the engine-room of a motor ship; there is evident advantage in extracting the oily vapors from the crankcase and condensing the lubricating oil, which represents a saving in the oil and prevents fumes extending to the engine-room. This arrangement is adopted in the *Asturias*. As each engine is built up of two groups (each group with an enclosed crank chamber), this involves the employment of two extraction circuits. There is a duct from each end of one section of the crank chamber (for four cylinders), and air is drawn through the crankcase into the duct by means of a motor-driven fan. It passes into what may be termed a condenser, although consisting of nothing more than a cooled iron cylinder. Any lubricating oil condensing is tapped off and led back to the sump. There are thus two condensers and two extraction fans to each engine, and it may be added that there is also a connection to the air-compressor crank chamber, so that there is little or no oily vapor entering the engine-room.

The cylinder lubricators are driven off the indicator shaft at the back of the engine and are arranged with eight feeds—three for the top and three for the bottom of the cylinder and two for the gland, this applying, of course, to each cylinder. At the ends are further lubricators for the bottom and top exhaust valves. The exhaust pipes for the upper and lower sections of the cylinders are taken to the forward end of the engine-room

and combine in one vertical pipe, being led thence into a large silencer in the after funnel of the ship.

At the front of each engine, not far from the control station, is a Siemens pyrometer, indicating the exhaust temperature from all cylinders, top and bottom. The average temperature when running at 110 R.P.M. was between 380 degrees C. and 400 degrees C., whilst the piston-cooling circulating pressure was 1 kilogram per square centimeter, or about 14.2 pounds per square inch, the pressure of the cylinder-cooling-water circuit being 1.8 kilogram per square centimeter, or about 25.5 pounds per square inch. The outlet temperature of the lubricating oil before entering the coolers was in the neighborhood of 110 degrees F. Among the interesting devices fitted in the engine-room may be noted a Bassler meter, by which the amount of fuel oil being used can be readily determined. A desoription of this apparatus was given in our issue of November, 1924. Pneumercator gauges of Kelvin, Bottomley and Baird's usual type are provided.

We understand that the engine-room staff will comprise in all 19 engineers and 23 greasers, which is, of course, a substantial reduction compared with the total number that would be required in a steamer, where it may be estimated that between 100 and 125 men would be needed.—"The British Motorship," March, 1926.

LAUNCHING ARRANGEMENTS OF H.M. SHIPS *NELSON* AND *RODNEY*.*

W. J. Berry, Director of Naval Construction.

Nelson and *Rodney* are the first ships to be designed and built to a standard displacement of 35,000 tons, the limit laid down for capital ships by the Washington Treaty on the Limitation of Armaments.

The keel plate of H.M.S. *Nelson* was laid at Messrs. Armstrong's naval yard, Walker-on-Tyne, on December 28, 1922, and that of H.M.S. *Rodney* at Messrs. Cammell Laird's shipyard, Birkenhead, on the same date. *Nelson* was launched on September 3, 1925, and *Rodney* on December 17, 1925. Both vessels were retained on the building slip until they were in a more advanced state of completion, for both hull and machinery, than is usual for battleships, and the armor was practically all on board; as a result the launching weight was approximately 20,000 tons, a very considerable figure in view of the length between perpendiculars being only

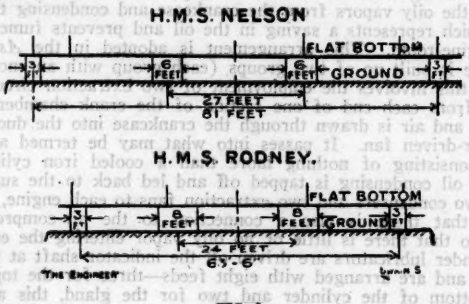


FIG. 1

* Abstract of paper read before Institute of Naval Architects, March 24, 1926.

660 feet. This launching weight was exceeded by about 2000 tons in H.M.S. *Hood* and by about 5000 tons in U.S.S. *Saratoga*, but in these ships the weight was carried on a much greater length.

Having regard to the launching weight and its distribution longitudinally, and more particularly, transversely, the side armor being on board, the builders proposed launching on four ways, the arrangements adopted in the two ships, after discussion with the Admiralty, being shown in Figure 1. General particulars are given in Table I.

Thorough investigations to estimate the condition of the ship at launch and the probable forces acting were made for two heights of tide, about 3 inches and 15 inches, respectively below the expected high tide, the lower tide being in the neighborhood of what was considered to be the minimum safe launching tide. The actual tide at the launch of H.M.S. *Nelson* was 2½ inches higher than the expected high tide, giving a height of water 13 feet 5½ inches over the way end. The actual tide at launch of *Rodney* was 1 inch below the lower of the two tides considered, the height of water over the way end being 12 feet.

TABLE I

	Nelson	*Rodney*
Length between perpendiculars, feet	660	660
Declivity of keel, inch per foot	½	0.503
Declivity of chord of ways, inch per foot	35/64	37.3/64
Camber of ways	18in. in 717ft.	24in. in 715ft.
Length of ground ways (inner), feet	717	715
Length of ground ways (outer), feet	533	521
Length of sliding ways (inner), feet	542	570.5
Length of sliding ways (outer), feet	245	216
Breadth of ways (each) (inner), feet	6	8
Breadth of ways (each) (outer), feet	3	3
Area of sliding surface, square feet	7,974	10,424
Ways apart outside to outside (inner), feet.	33	32
Ways apart outside to outside (outer), feet.	84	66.5
Fore edge of poppet from fore perpendicular, feet	76	65.6
Launching weight used in calculations (hull and contents only), tons	18,860	20,500
C. G. abaft amidships, feet	56.1	55
Water on way end	12ft. 1in.	12ft. 1in.
Poppet from way end when stern lifts, feet	189	198.5
Weight on poppet when stern lifts, tons	2,650	2,535
Weight on poppet at way end, tons	230	273
Mean initial pressure on grease, tons per square foot	2.435	1.97
Pressure on grease when stern lifts, tons per square foot, taken on a length of 16 ft.	13.8	9.9
Maximum load per foot run on way end during passage of mid-body, tons	119	156.6
Corresponding pressure on grease, tons per square foot	9.91	7.12
Least moment against tipping, tons-feet	280,000	270,000

After the launch the case of Nelson was again investigated in view of the actual conditions which obtained on the day of the launch. The principal results obtained from launching curves have been embodied in Tables I and II. In Table I. results are also given for *Rodney*, assuming height of water over way end to be 12 feet 1 inch. It will be noticed that the exceptionally high tide which gave 16½ inches more water over the way end

TABLE II

	Nelson
Launching weight used in calculations, tons. (Hull and contents, saddles, fore stopping and sliding ways)	19,780
C. G. abaft amidships, feet	55.4
Water on way end	13 ft. 5½ in.
Poppet from way end when stern lifts, feet	212.5
Weight on poppet when stern lifts, tons	2,580
Weight on poppet at way end, tons	Nil
Mean initial pressure on grease, tons per square foot	2.495
Pressure on grease when stern lifts, tons per square foot	13.44
Maximum load per foot run on way end during passage of mid-body, tons	66
Corresponding pressure on grease, tons per square foot	5.5
Least moment against tipping, tons-feet	405,000

than was assumed in the first calculation for *Nelson* very materially modified the results obtained.

An investigation was made to ascertain the maximum longitudinal bending moments and stresses that the ship might experience during her passage down the ways. For this purpose the lower of the two tides assumed in the calculations prior to launch was adopted, namely, a height of tide 12 feet 1 inch over way end.

A curve was drawn giving the longitudinal weight distribution of the ship. For the supporting forces the launch was assumed to be slow and the ship considered to be in equilibrium when at various positions on the ways. At each position the longitudinal distribution of the supporting force of buoyancy and pressure on the ways was calculated. Then by successive integration of the difference between the weight and supporting curves, a curve of bending moment acting on the ship at each position was obtained. The envelope of these curves was taken to represent the maximum bending moment that any section of the ship might experience. The principal results may be summarized as follows:

Hogging.—The maximum bending moment anticipated with tide line 12 feet 1 inch above way end was about 560,000 tons-feet, its position being somewhat abaft amidships. Adopting the usual method of calculating stresses, this gave:—Tensile stress in upper deck = 7.3 tons per square inch; and the compressive stress in outer bottom = 4.9 tons per square inch.

Sagging.—The maximum bending moment anticipated was about 210,000 tons-feet, its position being about 90 feet before amidships. At the position considered above for hogging the maximum sagging moment was 150,000 tons-feet, giving the following stresses at that section:—Compressive stress in upper deck = 1.1 tons per square inch; and the tensile stress in outer bottom = 1.5 tons per square inch.

The actual height of tide at launch of *Nelson* was 1 foot 4½ inches higher than the tide assumed in the original strength calculations, and the effect of this on the calculated results is as follows:

Hogging.—The maximum bending moment was reduced to 420,000 tons-feet, the corresponding stresses being: Tension in upper deck = 5.5 tons per square inch; and the compression in keel = 3.7 tons per square inch.

Sagging.—Except that the stern lifts after a shorter run, the sagging bending moments were not appreciably affected by the rise of tide. Hence the maximum calculated stresses in the upper deck at the position selected with the actual tide at launch became 5.5 tons per square inch tension and 1.1 tons per square inch compression. In view of the high estimated hogging stresses it was decided to take stress measurements during launch, using for this purpose, four in number, Fereday-Palmer strain recorders. The four

instruments were arranged in a line across the upper deck about 20 feet abaft amidships, two port and two starboard.

Launching Speeds.—Observations were made to enable a space-time curve of the ship to be plotted from which the velocity and acceleration of the ship during the travel down the ways were obtained. Curves for *Nelson* are shown on Figure 2 and are plotted to a base of distance run. The ship reached a maximum velocity of 18.8 feet per second after a run of about 320 feet, and the acceleration attained a maximum value of 0.8 feet per second per second after a very short run, and retained this value for a run of nearly 100 feet. The estimated coefficients of friction are 0.045 initial and 0.021 sliding. After passing the position of maximum velocity the retardation soon became practically constant at about 0.23 feet per second per second. Shortly after the fore poppet passed the after end of ways the retardation increased to about 0.55 feet per second per second; at this stage the first pair of drags had commenced to act.

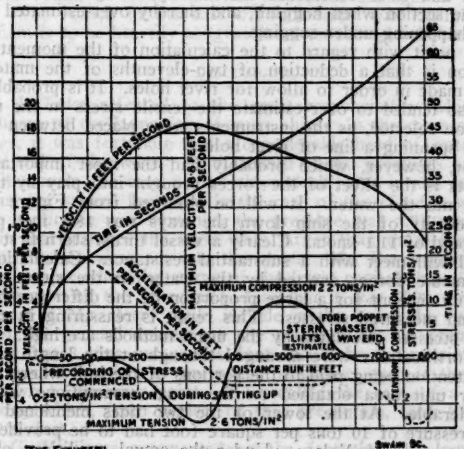


FIG. 2

At the launch of *Rodney* the acceleration was measured direct by means of an apparatus designed by Professor T. B. Abell, of Liverpool University. The results obtained were very similar to those obtained for *Nelson* by the usual methods.

The results obtained by the four strain recording instruments were in close agreement with one another. A mean of the four curves plotted to a base of distance run is given on Figure 2. On the same base are shown curves of time and velocity and acceleration of ship. The following points are noted:—

(1) Maximum tensile stress recorded = 2.6 tons per square inch, and occurs after a run of about 326 feet, at which point the position selected is still 117 feet from way end. (It was anticipated that the maximum stress would occur after a run of about 400 feet.)

This stress combined with the strength modulus of the section may be taken to represent a bending moment of 200,000 tons-feet instead of 420,000 tons-feet, the estimated maximum, *i. e.*, a reduction of 220,000 tons-feet.

(2) Maximum compressive stress recorded = 2.2 tons per square inch and occurs after a run of about 500 feet, at which point it was estimated the stern would lift at 13 feet 5½ inches tide.

This stress represents a bending moment of 300,000 tons-feet instead of 150,000 tons-feet, the estimated maximum, *i. e.*, a gain of 150,000 tons-feet.

In short, the maximum tensile stress recorded is only one-half of that estimated, whereas the maximum compressive stress recorded is double that estimated.

A factor that may have a bearing on this result is the method of taking account of armor and protective plating when estimating the inertia of the section. This was entirely neglected under tension but included under compression, and it is considered that this probably under-estimated the inertia of the section when hogging, and thereby over-estimated the stress, the reverse happening under sagging.

A further point with regard to the calculation of the moment of inertia of the section is that a deduction of two-elevenths of the material under tension was made in order to allow for rivet holes. It is probable that this deduction also tended to over-estimate the tensile stress in the upper deck at the position selected, as the instruments were placed between two frame stations, not spanning a line of rivet holes.

The factor, however, which probably had the most important bearing on the results is the effect of the forces brought into play by the passage of the ship into the water. It will be observed from Figure 2 that the maximum velocity of the ship down the ways was 18.8 feet per second, equal to a speed of 11.1 knots. Clearly a vessel thrust stern first into water at this speed will meet with a substantial resistance. Calculations showed that the dynamic forces exerted by the water on the stern of the ship during the run account for a large proportion of the difference between the calculated and measured stresses. This result is reassuring in cases where the hogging stresses calculated by the usual methods are high.

Internal fortifications were arranged throughout the length of the ship, particular attention being paid to the portions of the hull where the upward pressure per unit area obtained on the usual assumptions was expected to be considerable. At the lower of the two tides mentioned previously a possible pressure of 10 tons per square foot had to be provided for over the midship portion of *Nelson*. Under the actual conditions obtaining at launch 5½ tons per square foot was the estimated maximum pressure in this neighborhood. The pressure per square foot on the fore poppet when the stern lifted, assumed to be uniform over a length of 16 feet, was about 13½ tons, this figure being little influenced by the rise of tide.

Melted tallow to a thickness of about ½ inch was applied over the ground ways. Next a coat composed of three parts of tallow to one part oil was applied, following which were two coats composed of half tallow and half oil. Patches of soft soap at close intervals were then placed on the ground ways over the whole length of the sliding ways, and finally oil was sprinkled over the whole of the surfaces so treated. In addition, two coats of tallow were applied to the bottom surfaces of the sliding ways before these were pulled into position.

Owing to the restricted width of the river, it was necessary to ensure that *Nelson* should be brought to rest after a run of not more than 350 feet subsequent to the stem passing the after end of the ground ways. Eight drags were placed on each side of the ship, the total weight being 1162 tons. The first pair of drags was arranged to act when the stem of

the ship was still about 50 feet from the way end and the remaining drags came into operation in pairs after further runs of 25 feet between each pair. Actually the ship was brought to rest before the last pair of drags began to act.

No drags were necessary at the launch of *Rodney*, the width of the river being ample in this case.

For *Nelson* the holding, releasing and starting gear consisted of:—One dog-shore to each of the inner ways, two dog-shores to each of the outer ways, two 150-ton hydraulic holding and releasing triggers to each of the inner ways, two 100-ton hydraulic starting jacks to each of the inner ways, two 40-ton hydraulic starting jacks to each of the outer ways.

For *Rodney*:—One 200-ton and two 150-ton hydraulic holding and releasing triggers to each of the inner ways, one 100-ton hydraulic starting jack to each of the inner ways. On releasing the triggers both ships immediately started down the ways, without aid from the starting jacks.

The "breakage" of each ship in launching, both longitudinally and transversely, was found to be nil, and no structural damage of any sort was sustained. The launch in the case of both ships was entirely successful, and each firm is to be heartily congratulated on the splendid result obtained.

DISCUSSION.

Mr. W. J. Luke referred to the great responsibility which rested on those in charge of the launching of large ships. In the cases of the *Nelson* and *Rodney*, it was fortunate that a considerable head of water was available at the end of the ways. With regard to the pressures on the grease, he noted it was in the case of the ships dealt with in the paper 2.6 tons and 2.7 tons per square foot, and in the case of the *Hood* it was 2.45 tons. When the *Aquitania* was launched, the pressure was 2.27 tons per square foot, but in that instance the ways were larger than those from which the *Nelson* and *Rodney* were put into the water. Mr. Berry gave the weights on the fore poppet when the sterns of the latest capital ships lifted as 2600 tons; in the case of the *Hood* and *Aquitania* it was 5000 tons. It would be of interest if Mr. Berry could include in the paper the figures for the *Hood*. On the general subject of the pressure on the grease, they had been taught that 3 tons per square foot should be the maximum. It was important in cold weather to warm up the grease, even if the temperature was not down to freezing point. He recalled the case of a cargo vessel 400 feet long which, from neglect of that precaution, took fifteen minutes to move 100 feet.

Mr. H. G. Williams commented on the fact that the launches of the *Nelson* and *Rodney* were on four ways instead of two, which was a departure from precedent that could not be lightly undertaken. It was quite easy to make a sliding surface of any number of ways, but it was always open to doubt whether so satisfactory a distribution of the pressures would be obtained as with two ways only. The decision to employ four ways was made on account of the great beam of the ships. The chief risks in launching ships were, however, of what might be regarded as an accidental character, such as the failure of the tide to rise, the melting of the tallow on hot days, the sticking of the ship from other reasons, and the breaking of drag wires. A drag wire did actually break in one case, but no harm was done.

Professor Percy Hillhouse also referred to the pressure on the after end of the ways. The estimate was 9.9 tons, and the actual pressure 5.5 tons in one case and 7.12 tons per square foot in the other. In the *Transylvania* the figure was 11.2 tons; in the *Aorangi*, 21.3 tons; submarine *E-38*, 19.5 tons; *Renown*, 5.33 tons; and *Valiant*, 7.25 tons per square foot. A better figure than tons per square foot was tons per frame space. Mr.

Berry distributed the pressure on the fore poppet over a length of 16 feet. A more usual figure, he thought, was 30 feet. He would like more details of Professor Abell's speed-recording instrument.

Mr. E. L. Atwood said that the need for four ways launching was due to the excessive overhang of the vessels and the transverse distribution of weight, owing to the side armor being in position. It was an open question whether there was not an unequal distribution of pressure and a heavy demand on the outer ways, but it was thought that the weight would even out and prevent the pressure per square foot from being excessive at any point. He had an idea that the pressure on the outward ways might be enough to squeeze the grease out, but it was clear that the tallow would withstand pressures up to 7 tons per square foot without cracking. The information given in the paper was a valuable addition to knowledge concerning launching problems.

Professor T. B. Abell, in a written contribution, referred to the instrument used to measure the acceleration, which was of more importance than the distance run. It was shown that the stern of the ship would lift appreciably earlier than the estimated moment of lift, and in the case of the *Nelson* he thought it took place after traveling about 400 feet instead of 500 feet, as estimated. After the stern lifted, the changes in resistance from moment to moment would be very small, and the acceleration would become more and more uniform. The early lift of the stern involved an increase of pressure on the fore poppet beyond that estimated, and required a much larger reinforcement of the ground ways than would be derived from actual static calculation.

Mr. W. J. Berry, in reply, said that he would give the *Hood* figures for addition to the paper. In recent cruisers launched the pressure on the grease had been from 2.5 tons to 1.5 tons per square foot, but there was a little reluctance on the part of some of the ships to move off. The quality of the grease was not so good as before the war, and that fact must be taken into account in looking at the pressures recorded.—"Engineering," 26 March, 1926.

HIGH-PRESSURE WATER-TUBE BOILERS FOR MARINE PURPOSES.*

By Harold E. Yarrow, C. B. E.

The theoretical advantages of high steam pressures and temperatures in conjunction with turbine installations are well known and fully recognized. Sir Charles Parsons, in his paper on "Steam Turbines," given before the first World Power Conference in 1924, dealt with the subject in the following terms:—"By the use of higher steam pressures and temperatures, together with the lowest possible exhaust pressure and by other devices, it is expected in the future to reach an overall efficiency (from fuel to electricity) of 30 per cent, which is not inferior to the best results obtained by oil engines, even neglecting the cost of oil for cylinder lubrication."

Similar statements have been made by other responsible authorities, and naval architects and marine engineers are now beginning to consider seriously the application of high steam pressures and temperatures on board ship.

Sir John Biles, in his paper "Relative Commercial Efficiency of Internal-Combustion and Steam Engines for High-Speed Passenger Vessels," read before this Institution a year ago, gives an interesting comparison between the steam turbine and the internal-combustion engine, from which it is shown that the former, when high pressures and temperatures are employed, compares favorably in efficiency with the latter method of propulsion.

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* Paper read before the Institution of Naval Architects on Friday, March 26, 1926.

In submitting the present paper to the Institution, an endeavor has been made to investigate the practical application of high pressures and temperatures on board ship, and to make special reference to a definite step which has been taken in introducing high-pressure boilers in a passenger steamer at present under construction. The paper does not refer to any part of the machinery other than the boiler, nor has it been thought necessary to make any comparison in regard to weight, space occupied, etc., of high-pressure turbine installations as compared with internal-combustion engines or ordinary steam-pressure installations. This ground has been adequately covered by Sir John Biles in his paper, in which a very complete analysis will be found of the comparison between Diesel, steam, and what is termed H.P.T. (high-pressure turbine) installations.

The adoption of high pressures and temperatures on land has already made considerable progress, and, just in the same way as the introduction of the steam turbine for marine purposes followed some years after its introduction on land, so it would appear that history is repeating itself in the case of high-pressure boilers. Boiler installations having steam pressures of 500 pounds to 600 pounds per square inch and superheat temperatures of 700 degrees to 750 degrees F. are in commercial use in a number of power stations, and such pressures and temperatures may be said to have passed the experimental stage.

If, therefore, these pressures and temperatures can be adopted on land, there seems no reason why they should not likewise be adopted at sea. There are certain fundamental factors common to both, among them suitable materials and workmanship. The better and more uniform materials now in use and the improved metallurgical and manufacturing methods now adopted, remove from the designer's mind any anxiety as to materials. Copper, wrought-iron and cast-iron have been discarded in the manufacture of water-tube boilers for high pressures. Steel is in common use, and its uniformity leaves little to be desired. Certain of the non-ferrous alloys which are now used also contribute to that security which characterizes modern installations. Workmanship must, obviously, be of the very highest class throughout, just in the same way as the manufacture of the Diesel engine requires the greatest skill. It may therefore be accepted that, with good materials and workmanship, no trouble should be experienced in boilers working at the pressures and temperatures mentioned above.

There is, however, one difference between land and marine practice, namely, that in the latter case there is always a risk of a leaky condenser allowing salt to enter the system, although it must be remembered that there are many electric power stations where the condensing water is by no means entirely free from salt. The presence of salt can be detected immediately by various means, and, in the Navy, from the time a leaky condenser tube is discovered to the time when this tube has been plugged and the installation again under way occupies only a very short period. Sir Charles Parsons has proposed to sub-divide the condensers so that one of the condensers can be isolated if a leak is discovered. Both these methods are merely remedial, but with the investigations now being carried out, such, for instance, as adopting different materials for condenser tubes, it may be assumed that this trouble will be finally overcome. It is hardly conceivable, that, with the skill of the engineer and metallurgist, progress along any promising line of development will be retarded because no means can be found of keeping salt out of the feed system.

Even under existing conditions, Atlantic liners, cross-Channel boats, and other types of steamers, as well as vessels of the different navies of the world, are running satisfactorily with water-tube boilers.

Fear has been expressed that, with high temperatures and increased pressures, there will be a difficulty in keeping steam joints tight, and that

this will be more serious on board a vessel than in a power station, as the latter generally consists of a large open building and all joints are easily accessible. There does not seem any valid reason to fear this difficulty, for, if proper materials and high-class workmanship are employed, steam joints can be made quite satisfactory. A suitably designed metal-to-metal joint with strong bolts and thick flanges should give no trouble. In one respect the problem is simplified in a marine installation as compared with land boilers, because in the latter case the units are usually much larger and the valves and pipes are, in consequence, very much bigger than on board ship. For instance, the steam pipe from each of the Yarrow boilers at the County of London Electric Supply Company's station at Barking, working at 400 pounds per square inch and 725 degrees F. superheat, has a diameter of 8 inches and joins up to the 12-inch main steam line. In a steamship of 24,000 shaft horsepower, having six boilers of 575 pounds pressure and 750 degrees superheat, the diameter of the main steam pipe from each of the boilers would be only 4 inches and would connect into two 7-inch steam pipes in the engine room. The valves and boiler fittings would be of smaller size and their design therefore simplified.

In considering the problem of turbine installations with high-pressure boilers, it may be assumed that a boiler efficiency of not less than 85 per cent will be realized when air heaters are fitted, and, in the case of boilers with straight and well inclined tubes, the rapid circulation of water keeps the tubes clean, and therefore a high efficiency can be maintained over long periods of service.

It may be of interest to the meeting to give some particulars of two high-pressure boilers, coal fired, which will be installed on a passenger vessel now being built by Messrs. Wm. Denny and Brothers, for Turbine Steamers, Limited, the managing owners of which are Messrs. John Williamson and Co. In determining the type of boiler for this vessel, it was necessary to select a design that would be suitable for general mercantile practice and one preferably free from as many bolted and riveted joints as possible due to the high pressure to be adopted. It was also considered important that all the pressure parts should be, as far as possible, circular in section. After careful investigation it was decided to adopt the Yarrow type, and the order was placed for the two boilers by the main contractors, the Parsons Marine Steam Turbine Company, with Messrs. Yarrow and Co.

The design of the boiler has been passed by the Board of Trade, whose officers have offered many valuable suggestions and who have given every assistance to facilitate this development. Each boiler, the general arrangement of which is shown in Figure 1, has a total heating surface of 3420 square feet, of which the superheating surface is 870 square feet; in addition there are 2200 square feet of air-heating surface for pre-heating the air prior to its admission to the closed ashpit. The safety valves are set to 575 pounds pressure and the final temperature will be from 700 degrees F. to 750 degrees F. The test pressure is 913 pounds. The design generally, while not unlike that used in the Navy, has been based on mercantile practice and more closely follows the design of the Yarrow land-type boiler now in use at several electric power stations.

As will be seen from Figure 1, the boiler consists of a forged steam drum connected to the three forged water drums by means of straight tubes expanded and bell-mouthed in accordance with the usual practice. Between the two water drums on the right-hand side is a superheater, which consists of a forged drum with a number of U-tubes and is placed between the two generating elements. The gases all pass up on one side of the boiler through the air-heater situated above the boiler to the funnel. The reason for adopting a single-flow type of boiler, where all the gases pass through one side, is because this design makes, in this particular case, a somewhat better

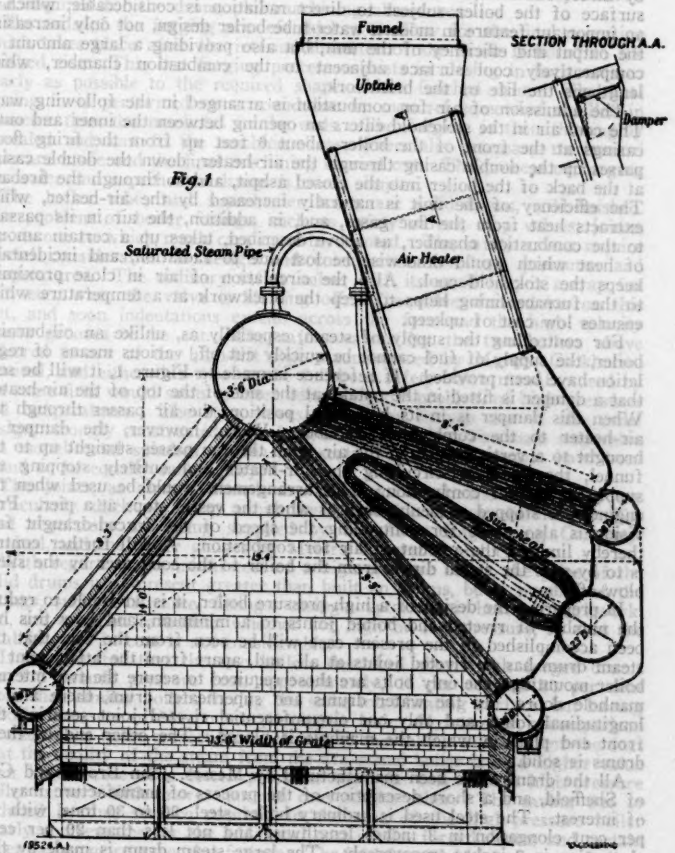

Funnel
SECTION THROUGH A.A.
Uptake
Damper
Saturated Steam Pipe
Air Heater
3'6 Dia.
8'4
15'4
33'6 Width of Grate

Fig. 1

arrangement in the ship than the double-flow type, where the gases pass equally through each side. The generating element on the left-hand side of the boiler absorbs a considerable portion of the heat from the furnace by direct radiation, and it will be noted that the proportion of the total surface of the boiler subject to direct radiation is considerable, which is an important feature in modern water-tube boiler design, not only increasing the output and efficiency of the unit, but also providing a large amount of comparatively cool surface adjacent to the combustion chamber, which lengthens the life of the brickwork.

The admission of air for combustion is arranged in the following way: The cool air in the stokehold enters an opening between the inner and outer casings at the front of the boiler, about 6 feet up from the firing floor, passes up the double casing through the air-heater, down the double casing at the back of the boiler into the closed ashpit, and so through the firebars. The efficiency of the unit is naturally increased by the air-heater, which extracts heat from the flue gases, and, in addition, the air in its passage to the combustion chamber, as above described, takes up a certain amount of heat which would otherwise be lost due to radiation, and incidentally keeps the stokehold cool. Also the circulation of air in close proximity to the furnace lining helps to keep the brickwork at a temperature which ensures low cost of upkeep.

For controlling the supply of steam, especially as, unlike an oil-burning boiler, the supply of fuel cannot be quickly cut off, various means of regulation have been provided. If reference is made to Figure 1, it will be seen that a damper is fitted in the uptake at the side of the top of the air-heater. When this damper is in its horizontal position, the air passes through the air-heater to the combustion chamber. When, however, the damper is brought to a vertical position the air from the fan passes straight up to the funnel, thereby short-circuiting the air-heater and entirely stopping the supply of air for combustion. This arrangement would be used when the engines are stopped suddenly, such as when the vessel stops at a pier. Provision is also made for controlling the speed of the forced-draught fan, thereby limiting the amount of air for combustion. A still further control is to by-pass the steam direct from the boiler to the condenser by the silent blow-off.

In preparing the design of a high-pressure boiler, it is advisable to reduce the number of riveted and bolted joints to a minimum, and how this has been accomplished in the present case will be seen from the fact that the steam drum has no riveted joints at all, and, apart from the attachment of boiler mountings, the only bolts are those required to secure the two internal manhole doors. In the water drums and superheater drum, there are no longitudinal joints and only one circumferential riveted joint securing the front end plate in which the manhole is placed. The other end of these drums is solid.

All the drums have been manufactured by Messrs. John Brown and Co., of Sheffield, and a short description of the process of manufacture may be of interest. The steel used is ordinary boiler steel, 26 to 30 tons, with 30 per cent elongation in 3 inches lengthwise and not less than 20 per cent elongation in 3 inches transversely. The large steam drum is made by the following method:—A large octagonal ingot is forged under a hydraulic forging press to a round billet, it is then cut to a length which will give the weight necessary for the finished forging and is carefully examined for any surface defects. The billet is then bored throughout its entire length with a hole large enough to take a mandrel for the second forging operation.

The second forging operation consists in expanding the billet to a larger diameter without increasing the length; the internal diameter after this operation is slightly less than the finished diameter of the steam drum. The expanded forging is reheated ready for the third operation, which con-

sists in forging on a mandrel in such a manner as to increase the length without increasing the diameter. A mandrel is used of a diameter approximately equal to that of the finished drum, and forging is continued until the required length is obtained. The thickness of the main body is now that required for the drum, plus machining allowances.

The ends of the forging are left thicker to allow for the final operation which takes place after the internal diameter of the drum is bored to the finished size. This last forging process consists in closing in both ends as nearly as possible to the required shape of the finished drum. The final closing in is performed on a round bar or mandrel somewhat smaller in diameter than the manholes in the ends. The forging is then thoroughly annealed and sent to the machine shops, where the necessary mechanical tests are made and the final machining carried out. The process of forging the water and superheater drums is much simpler.

The boiler tubes and superheater tubes are solid-drawn, cold-finished. The problem of boiler fittings, such as safety valves, stop valves, joints, etc., is one to which much care must be devoted. The safety valves require special consideration in design and construction for high pressures and temperatures. The ordinary safety valve, with its small lift, is a trap for grit, etc., when the valve is blowing off. The valve shuts down on this grit, and soon indentations extend across the face and the high velocity causes grooves to be cut. The safety valves for the two boilers above referred to are of the high-lift type manufactured by Messrs. Cockburns, Limited, and either this type or the full-bore safety valve manufactured by the same firm would seem to meet adequately the problem of high pressures and temperatures. The stop valves and other fittings are of the usual design, strongly constructed and of suitable materials. The main stop valve has monel-metal spindles, valves and seats, but other materials have been found to be equally satisfactory.

The joints, as already stated, are metal to metal, and the flanges are particularly thick so as to avoid the possibility of any distortion. The feed regulators are being supplied by the Aster Engineering Company, Limited, and the water gauges by Messrs. Dewrance and Co.

With regard to the initial cost of a high-pressure boiler, the price of the solid drums is at present greater than built up drums, but a number of the fittings, such as furnace, fittings, casings, uptakes, air-heaters, brickwork, etc., are common to boilers whether for high or low pressure, so that the total increase in cost is not considerable.

It is estimated that the two boilers described above with forged drums for 575 pounds pressure cost, approximately, 15 per cent to 20 per cent more than similar boilers having the same total heating surface, but having a boiler pressure of 250 pounds per square inch. If, however, a comparison is made on a more correct basis, that is to say taking into account the fact that the quality of the steam from the 575-pound boiler is of a higher value than in the case of a lower pressure, then the cost of the high-pressure boiler is very little more than the cost of the other. The saving due to reduced heating surface required in the case of the high-pressure boiler nearly compensates for the additional cost of the forged drums and certain other rather more expensive fittings. The initial cost of high-pressure boilers is therefore no handicap to their introduction.

Figure 2 is a section of a Yarrow boiler suitable for 1000 pounds pressure, the boiler being of sufficient size to produce steam for 4000 S.H.P. when being worked at a very moderate rating. This boiler is of the double-flow type, but it can, of course, be designed for single flow, depending upon the general lay-out of the boiler room.

Although the actual trial results of the vessel for which the two 575 pounds boilers are being constructed are not yet available, it was thought, nevertheless, that a reference to the boilers might be of interest.

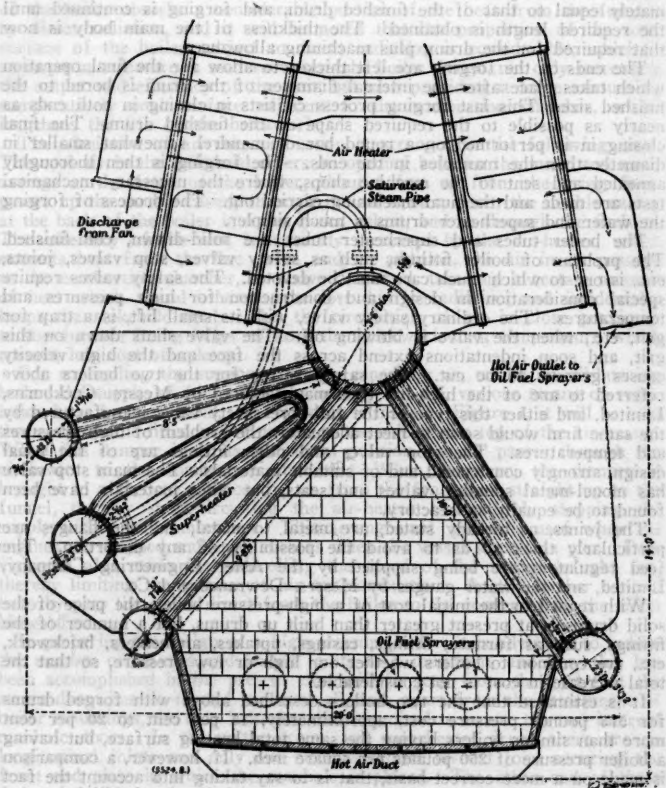


FIG. 2.—YARROW MARINE BOILER FOR 1000 POUNDS PRESSURE.

The paper read by Sir John Biles before this Institution a year ago pointed out the possibilities of high-pressure steam installations for marine purposes. This paper describes a definite step that has been taken in introducing extra high-pressure boilers on board ship, and it is to be hoped that a paper may be forthcoming when the trials of the vessel have been completed, the results of which will be awaited with keen interest.—"Engineering," April 2, 1926.

SOME RECENT MODIFICATIONS TO WATER-TUBE BOILERS OF THE THREE-DRUM TYPE FITTED IN H.M. NAVY.*

By Engineer-Captain L. M. Hobbs, R. N.

The general design of the three-drum type of water-tube boiler, often associated with the name of Yarrow, is so well known that no description of it is necessary in this paper. As a result of experience in H.M. ships during and after the war, certain modifications have, however, been made to details of this type of boiler, and it is thought that a brief description of these alterations may be of interest, and may be instrumental in preventing in the future a recurrence of the features which have led to some of the difficulties experienced.

The steam drums of practically all Yarrow type boilers fitted in H.M. ships during recent years have for convenience in manufacture had a standard internal diameter of 50 inches with a wrapper plate five-eights of an inch thick, double butt strapped on to the tube plate. Experience has shown that grooving of the wrapper plates developing into cracks is likely to occur along the upper edge of the internal butt straps of such drums, especially where these seams have been heavily caulked. To prevent this, caulking the edges of such internal butt straps is now prohibited, as the stress in the material and the initial nicking of the plate, caused by the caulking, is peculiarly liable to set up corrosion of the plates. It has also been found advisable to bevel the upper edges of internal butt straps to an angle of 75 degrees to prevent moisture lodging on the plate edge for any considerable period when the boiler is emptied. Other changes that call for comment in connection with steam drums are the omission of external downcomer pipes during the early years of the war, and the recent elimination of the scum pan and its fittings, which have been found of little use on service, where pure feed-water is available.

The steam drum end-plates, more particularly when fitted to accommodate downcomer tubes, were found to develop grooving, which resulted in the cracking of the plates in the root of the flange. Such plates are now pressed in one operation and should not be locally heated during any stage of their manufacture. The omission of downcomers has enabled the length of the boiler drums to be reduced by at least 6 inches, and has had no apparent effect on the efficiency of the water circulation, while the saving of weight has been of the order of 2 per cent for boilers of the same power. Where economy of weight and space is not a primary consideration, the retention of downcomers probably ensures a more definite circulation throughout the boiler.

Previous to the year 1914 water drums of boilers of the Yarrow type were generally of oval section, although in a few cases, as in the Normand boiler, circular barrels were fitted. In 1913, while water testing the boilers of two destroyers, the plates of the lower drums developed horizontal cracks about 5 to 6 feet in length along the seam connecting the wrapper and tube plates. In 1914 similar cracks were observed in the water drums of a number of other vessels, the ages of whose boilers were in some instances as much as 14 years and in others as little as 5 years. Investigation led to the opinion that the chief cause of the trouble experienced was the sharp edge left on the plate by the caulking tool, combined with the racking of the seams. It was noticed that the defects referred to occurred chiefly in the boilers fitted in destroyers and in the smaller cruisers with a quick roll, while Yarrow boilers in large cruisers and battleships were to a large extent immune from serious trouble.

* Abstract of paper read at the Spring Meetings of the Sixty-seventh Session of the Institution of Naval Architects, March 26.

Experiments were carried out to ascertain if it were possible to stiffen water drums of this shape sufficiently to prevent distortion taking place under normal pressure by the use of stays and struts.

It was, however, found practically impossible to prevent a very appreciable change of forms, and any idea of stiffening by struts and stays was abandoned. It was therefore decided that all new boilers should be fitted with water drums of circular cross-section with double butt-strapped joints. To meet the case of existing boilers, it was arranged to fit cover straps over those lap joints where defects were suspected, and by this means a large number of boilers were kept safely and efficiently in use throughout the war.

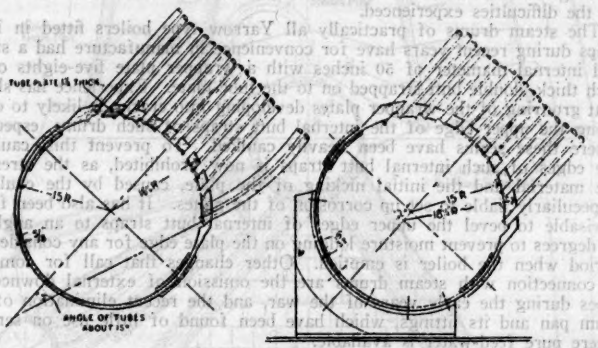


FIG. 1.—STANDARD CIRCULAR WATER DRUM AND, ON RIGHT, PROPOSED NEW FORM.

The standard size of circular water drum now used in H.M. service is one having an internal diameter of 30 inches, with tube plates 1 1/16 inches thick and wrapper plates 5/8 inch thick, as shown in the left-hand drawing, Figure 1. This provides convenient space for internal examination, and for a sufficient number of tube rows in the tube plate, while maintaining a minimum length of 3/4 inch for the expanded portion of the wing tubes in the tube plate. It is obvious the expanded portion of the more central tubes is unnecessarily great, and the design of the tube plate, shown on the right in Figure 1, was proposed, in which the inside of the drum was circular while the outside of the tube plate was machined and made thinner at its center. By this method the weight of the tube plate is reduced, but a considerable amount of machining is involved.

In the older form of water drum, cracks between the tubes holes have been frequently observed on the interior of the tube plate; and they are still sometimes experienced with circular drums. They are usually due to the size of the arbored holes provided to take the belled ends of the tubes. If arboring is excessive, a piece of metal of knife-edge thickness is left between the holes for the fire rows. Any racking stress on the plate is almost bound to start a crack at such a point, and any sharp edges should originally be rounded by grinding with an emery wheel, or preferably the arboring should be of rather smaller diameter, thus leaving more metal between adjacent holes.

TUBES.

In the earlier type of Yarrow boiler the tubes were all straight, but it was found that after steaming the two fire rows usually distorted. It was therefore arranged to give an initial bend to these two rows, and thus avoid further distortion. When the circular water drum was introduced, the smaller radius of curvature of the tube plate reduced the number of rows of straight tubes that could be rolled into the plate. To retain the same number of tube rows as had been fitted with oval pockets it became necessary that the fire rows should enter the lower drums at an angle to the remainder if sufficient rolling surface was to be provided for the tube ends, and for some years the type of bend adopted has been that indicated at (A) in Figure 2. The radii of the bends is not less than thirty times the external diameter of the tube, as this is the minimum radius to which tubes are allowed to be bent without further annealing. A straight piece of tube 3 inches long should also be left outside each tube plate before the bend commences, so that there is no likelihood of the tube expander biting into the tube unevenly.

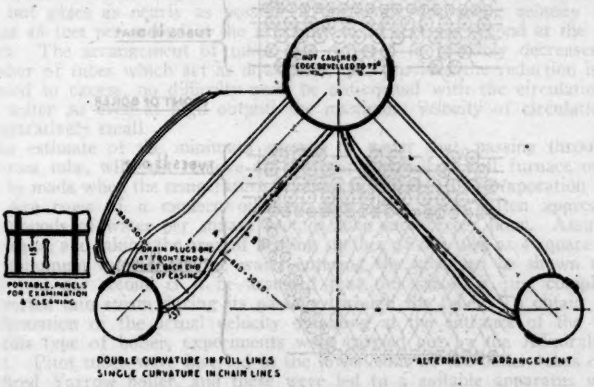

PORTABLE PANELS FOR EXAMINATION & CLEANING
DOUBLE CURVATURE IN FULL LINES
SINGLE CURVATURE IN CHAIN LINES
ALTERNATIVE ARRANGEMENT

FIG. 2.—ARRANGEMENT OF FIRE ROW TUBES.

It has, however, been found that the arrangement shown at (B) in Figure 2, where the fire row tubes have a single curvature only, is more satisfactory than the previous method. In the first case, if the tube does distort it is liable to deflect at either bend, and if adjacent tubes distort in opposite directions the bunching and lack of alignment becomes very marked. In case (B), if the tubes distort there is a greater tendency for them all to set in the same direction, and when boilers are re-tubed this method of fitting the fire row tubes is now generally adopted.

About the years 1919–20 the boilers of a number of vessels were constructed with a special arrangement of tubes designed to save weight and reduce the heat stress on the fire row tubes while maintaining the same boiler efficiency. This arrangement is shown in Figure 3. With the usual arrangement of tubes, the two fire rows exposed to radiant heat obviously form the most effective portion of the evaporative surface of the boiler; in fact, it is estimated that more than 40 per cent of the total heat given to such a boiler at full power is absorbed by the first fire row alone.

With this later arrangement of fire rows it was considered that a larger tube surface would be exposed to the maximum furnace temperature with a corresponding improvement in the total evaporation.

Moreover, when hot gases flow over a cooler surface the heat given up by conduction for a given temperature difference is a function of the velocity with which the gases flow over the heated surface. Decreasing the number of tubes in the fire rows as shown reduces the speed of flow of the furnace gases as they pass between the fire rows, and thus less conducted heat is

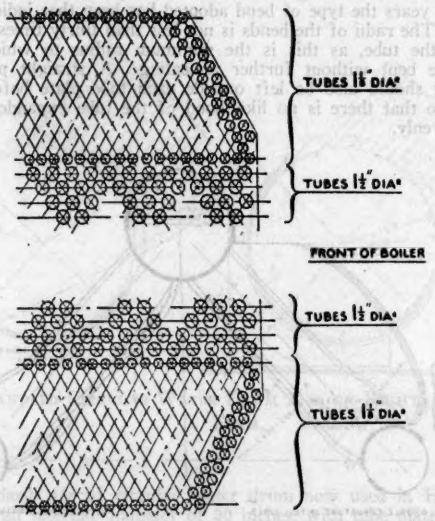


FIG. 3.—IMPROVED ARRANGEMENT OF FIRST ROW TUBES, ADOPTED 1919–20.

given to the first tubes and a greater proportion to later rows. To carry this idea to its logical conclusion it was desirable to reduce the total area between the tubes for the flow of gases in proportion to the decrease of temperature and volume of these gases. The volume of the gases at the wing row is about two-thirds of their volume at the first row of small tubes, since the volume varies as the absolute temperatures at these points, which experiment has shown are about 1250 degrees F. and 1850 degrees F. absolute respectively under full power conditions. As indicated in the figure, this reduction in area is carried out by omitting six wing row tubes at each end of the boiler and sloping the front and back casings as shown. By this means the area is reduced by about one-sixth, the remaining reduction of one-sixth being obtained by a shallow baffle hanging from the steam collector and extending the length of the boiler. This design keeps the velocity of the gases past all the tubes more nearly constant, eliminates a number of tubes in which the evaporation is comparatively small, and saves weight without appreciably affecting the output or efficiency of the boiler. The net result of the alteration in practice appears to be a saving of nearly 5 per cent in the weight

of each boiler, and a reduction of about 14 per cent in the tube surface compared with boilers of similar power fitted with the older arrangement of tubes. While no difficulty has been experienced in obtaining the same maximum power from boilers with this reduced heating surface, there is an indication that the tubes at the apex of the V lanes nearest the fire are more liable to distort than with the more usual and regular arrangement of fire row tubes. This method of fitting the fire row tubes has now been discontinued, but the arrangement of wing rows is retained and the larger spacing between the fire rows can, if desired, be obtained by bending the tubes with a single curve as previously referred to and turning alternate tubes upside down. This method is indicated on the right-hand side of Figure 2, and any area desired within limits can be arranged for the flow of gases by suitably adjusting the bends between adjacent tubes. Obviously, if the area for flow be large the frictional resistance to the flow of the gases will be small, but the transfer of heat to the tubes may be inefficient; while if the area be made too small the higher gas velocity will give a better heat transfer, but the restricted passages may unduly increase the work to be done by the fans. The areas as generally arranged give the hot gases as nearly as possible a uniformly decreasing velocity from about 45 feet per second at the fire rows to 35 feet per second at the wing rows. The arrangement of tubes just referred to probably decreases the number of tubes which act as downcasts, but, provided the reduction is not carried to excess, no difficulty need be anticipated with the circulation of the water ,as even at high outputs the maximum velocity of circulation is comparatively small.

An estimate of the minimum amount of water that, passing through a fire-row tube, will just remove all the heat supplied at full furnace output can be made when the temperature gradient is known. The evaporation from the fire rows of a modern oil-fired destroyer's boiler often approaches 100 pounds of water per square foot of tube surface per hour. Assuming this value and taking the area of heating surface of one tube as 4 square feet, the minimum velocity of the water entering the tube can be shown to be 0.22 feet per second if it be assumed that this water is just completely converted into steam during its passage through the tube. To obtain some confirmation of the actual velocity obtaining at the entrance of the tubes of this type of boiler, experiments were carried out by the Admiralty in 1921. Pitot tubes were inserted in the lower ends of the boiler tubes of an oil-fired Yarrow boiler, and these were led to a suitable apparatus which measured the velocity of the stream entering the tubes. Although too much weight must not be given to the quantitative results obtained, the indications were that the velocity of flow through a 1½-inch diameter fire-row tube increased as the rate of burning of the oil increased, up to about 0.5 pound of oil burnt per square foot of heating surface per hour, but after this up to a rate of burning of 1.1 pound per square foot per hour the velocity of the water remained steady and had a maximum value of 2.92 feet per second. In the second fire row the maximum water velocity was 1.5 feet per second when burning oil at a rate of about 0.6 pound per square foot of tube surface per hour, but this velocity fell off as the rate of forcing increased until at full power (approximately 1.2 pounds of oil per square foot of heating surface per hour) the apparent velocity did not exceed 0.9 foot per second. The apparatus was also tried in the fourth row of 1½-inch diameter tubes. In this case the readings obtained were somewhat erratic, the maximum velocity measured when burning oil at a mean rate of 1.2 pounds per square foot of heating surface being 0.65 foot per second, but this velocity was also obtained at lower rates of evaporation, while on two occasions reversals of flow were recorded.

The direction of flow through the wing tubes is intimately connected with the boiler-feed arrangements. In earlier designs it was usual to supply the main feed direct to the water pockets and the auxiliary feed to the steam drums, and the main feed actually entered separate compartments in the wings of the water pockets, which were baffled off from the remainder of the pocket. The last three rows of tubes entered these feed compartments, and the intention was that these tubes should act as feed heaters. Experience, however, soon showed that rapid pitting and corrosion occurred at the lower ends of these tubes, and that both the baffle and the riveted joint were rapidly eaten away. This was undoubtedly due to the air given off from the feed water as it entered the comparatively hot drum. Moreover, the spasmodic entrance of cold feed had a detrimental effect on the riveting of the drum joint, and leakage from these seams was very prevalent.

In view of these unsatisfactory features, all such feed baffles have been removed, and it is now considered more satisfactory to take both the main and auxiliary feed connections to the steam drum, where any air given off passes over with the steam in a less nascent condition. The feed is now arranged to enter the drums through a long internal perforated pipe with its end blanked off. The perforations consist of three or four lines of ¼-inch or 5/16-inch holes which discharge the feed upwards at an angle to the surface of the water in the drum, and the total area of these holes should not be less than twice the area of cross-section of the feed pipe. In earlier designs, where a plain open-ended, internal feed pipe was fitted, it was found that considerable corrosion of tube ends and tube plates occurred in the vicinity of the open end where the feed water was discharged into the drum.

DISTORTION OF TUBES.

Distortion of boiler tubes, if not due to overheating, is usually attributable to insufficient provision for the expansion of the tubes. This may result from improper adjustment of steam-drum stays or boiler feet. The movement of boilers in a ship is necessarily limited in extent to ensure their stability in a seaway, but securing stays attached to steam drums should give sufficient freedom for any vertical expansion that may occur when the boilers are steamed, and it is important to ensure not only that the original designed clearances are provided, but that they are maintained on service. In a modern boiler of the type under consideration the load on each foot may exceed 12 tons, and it is of great importance that efficient arrangements are provided for the free expansion of the boiler feet. Many boilers have their feet secured to the bearers by bolts passing through oval or enlarged holes in the feet. By suitable collars the nuts of the securing bolts are prevented from being screwed down hard except in the case of one foot which is definitely fixed, but without removing the nuts and washers it is difficult to ensure that the designed clearances are maintained in the right direction. A much more satisfactory method is that in which the boiler feet are constrained to move in guides similar to those fitted to the sliding feet of turbines, and this arrangement is now being generally adopted. By this means unlimited expansion in a fore-and-aft direction is provided, while the athwartship expansion is limited to ¼ inch on the outer side of each foot. Stops are fitted to the front feet in such cases to prevent the whole boiler sliding in a fore-and-aft direction. It is desirable that provision should be made for lubrication by suitable holes and oil grooves in the sliding feet.

A further modification found necessary during the last few years is the greater portability of the front and side boiler casings owing to the somewhat restricted spaces in the lower corners of the wing casings of boilers of the Yarrow type. The lower ends of the wing tubes are not easy of access, and in some cases where damp had descended from funnels or leaky stop-

valves a number of tubes have been found nearly perforated, due to a damp coating of soot which had remained for a considerable period without removal. To provide for easier examination and preservation of such parts, the wing casings of all boilers of this type are now having portable panels fitted to them, while drain plugs are also being provided to remove any moisture that may accumulate. This alteration is indicated in Figure 2, and should prevent a recurrence of the difficulty experienced.

The foregoing represent some of the changes that have been found desirable in this type of boiler during recent years, but references to any arrangements under trial and still in the experimental stage have been omitted.

DISCUSSION.

Vice-Admiral Sir George Goodwin said that although the paper purported to deal with "recent" modifications, some of those mentioned were 10 years old. While the author had rightly said that the omission of down carrier tubes on the Yarrow and White-Forster boilers had had no apparent effect on the efficiency of water circulation, this should not be taken to mean that down-carrier tubes could safely be omitted from all boilers. There was a good deal more to be said with regard to the difficulties experienced with water drums, although the paper dealt with the matter at fair length, and he hoped that fuller details of the many interesting and instructive experiments that were made at the time these difficulties were being experienced would be forthcoming.

Mr. W. W. Marriner said it was satisfactory to know that there had been so little trouble with the Yarrow boilers, although boilers of that type constituted 95 per cent of the total power used afloat by the Navy in the war. In the development of the Yarrow boiler the Admiralty had always given every encouragement, and from the paper it would seem that most of the troubles experienced had been due either to the nature of the steel or the bad treatment it had received during the manufacture of the boiler.

The Author, replying to the discussion, said the paper was not controversial, but it contained all the information which he was authorized to give, by the Admiralty, at the present time. That would explain Sir George Goodwin's comment on the use of the word "recent" in the title.—"Shipbuilding and Shipping Record," April 1, 1926.

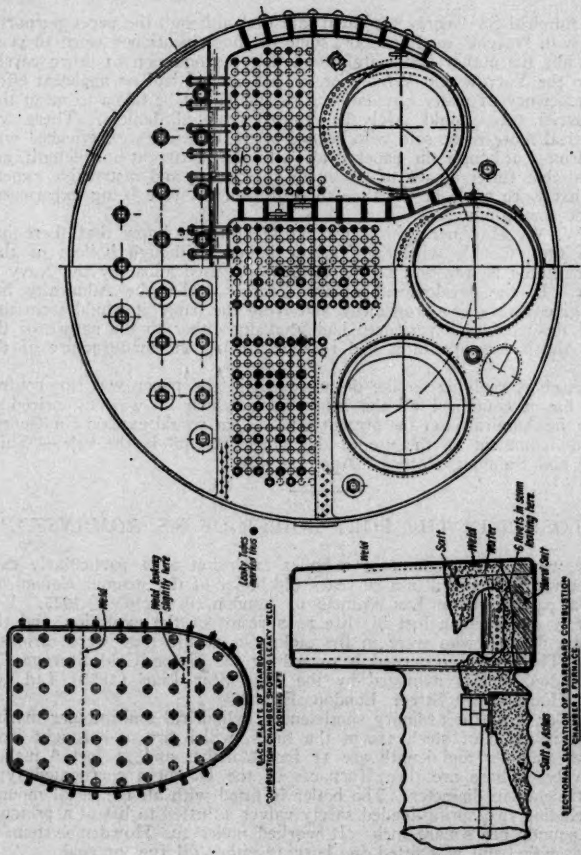
EXPLOSION IN THE PORT BOILER OF S.S. *ROMANSTAR*

Below we give particulars of a boiler explosion of a particularly rare character which occurred in a 30-years' old boiler of the steamer *Romanstar* during a passage from Las Palmas to London, in October, 1925. Unfortunately one fireman lost his life as a result of the explosion, and the other two firemen who were in the stokehold at the time were seriously injured. The vessel concerned is owned by the Union Cold Storage Co. Ltd., London, and is managed by the Blue Star Line (1920) Ltd., of Holland House, Bury Street, London, E.C. 3.

The boiler is of the ordinary single-ended cylindrical multitubular marine type and is made of steel, except the tubes, which are of wrought iron. Its mean diameter and length are 14 feet 3 inches and 11 feet 6 inches, respectively. There are three furnaces of the Deighton corrugated type, 3 feet 7 inches in diameter. The boiler is fitted with all the usual mountings, including two spring-loaded safety valves adjusted to lift at a pressure of 160 pounds per square inch. It worked under the Howden system of forced draught, and was fitted for burning either oil fuel or coal.

Considerable repairs have been carried out on this boiler from time to time, and between April and August of 1917 all three furnaces, all plain smoke tubes, large portions of the combustion chambers and a large number of the combustion chamber screwed stays were renewed. These repairs were carried out to the satisfaction of Surveyors to Lloyd's Register of Shipping, and, on completion, the boiler was tested by hydraulic pressure to 240 pounds per square inch.

The boiler has been periodically inspected by Surveyors to Lloyd's Register of Shipping, and was last inspected by one of their Surveyors at Liverpool in January, 1925. It was also examined by Mr. J. R. Douglas, Superintendent Engineer, early in July last.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A salt chamber, which formed in the bottom of the starboard combustion chamber of the port boiler by the evaporation of leakage of boiler water containing salt in solution, exploded with considerable violence. The smoke-box door was blown open, all the furnace firebars fell to the bottom of the furnace, and practically all the burning coal which was in the furnace and the heated contents of the smoke tubes were discharged into the stokehold. Fireman Wood, who was apparently in the act of working this fire when the explosion occurred, received such injuires that he died three hours later without regaining consciousness. The two other firemen, Lynch and Davies, who were on duty in the stokehold at the time, were seriously injured by being burnt.

No part of the boiler itself failed, and it continued to hold its water level and steam pressure during the passage of the vessel from the position where the explosion occurred until her arrival at Brest, a distance of about 70 miles.

The explosion was caused by the rupture of a salt chamber. Having regard to the violence of the explosion, this chamber, in the opinion of the Board of Trade Inspector, contained water of approximately the same temperature and pressure as that in the boiler from which it had leaked, and the instantaneous conversion of a large proportion of this water into steam, consequent upon the sudden release of pressure when the crown of the salt chamber ruptured, resulted in an explosion.

The S.S. *Romanstar,* ex *Brodmead,* ex *Rakaia,* is a steel single-screw cargo vessel of 5616 tons gross, and is classed + 100 A1, 7.25, and + L.M.C. 1.25, in Lloyd's Register of Shipping. She is fitted with triple-expansion, surface condensing engines of about 2300 indicated horsepower. The vessel is insulated throughout below the main deck for the carriage of refrigerated cargo, and two large Linde type ammonia refrigerating machines are installed in the main engine room and situated one on each side of the main engines on the bottom platform. Steam for all purposes is supplied by three main boilers of the same design, previously described. These are ranged athwartships, side by side, and fired from the forward end. The stokehold, which is entered from the engine room via a passage between the center and starboard boilers, is somewhat restricted, as the bulkhead forming the forward boundary is only 7 feet from the furnace fronts.

According to the evidence, when the vessel left Liverpool on July 21 last she was well found, and her machinery and boilers were in a satisfactory condition. The voyage to Durban, Port Elizabeth, Cape Town and Las Palmas was uneventful, and all went well except that it was difficult to maintain anything approaching full pressure steam on the boilers.

The vessel was at Cape Town from August 28 until September 9, and during that time the combustion chambers and smoke tubes of all boilers were cleaned as far as possible. It might here be mentioned, however, that owing to the seamen's strike the vessel had to be ready to sail at any moment during the last six or seven days of that time with a very valuable cargo consisting of about 100,000 cases of oranges.

When the starboard combustion chamber of the port boiler was then being cleaned there was a considerable amount of mixed salt and ashes in the bottom of the chamber which had formed solid and which was only partly cleaned out on account of the difficulty of its removal, and little importance was attached to its presence as there were no signs of leakage then observed in the lower portion of the combustion chamber.

The vessel arrived at Las Palmas on September 30, and after taking on board bunker coal and fresh water proceeded to London on the same day. All went well until 10.20 P.M. on October 6, when an explosion occurred in the starboard combustion chamber of the port boiler. Mr. John Coombe, the chief engineer, was standing on the bottom platform of the engine room at this time and heard a dull heavy thud which was accompanied by a

Port Boiler of SS. "Romanstar," showing Location of Salt Chamber and Leaky Tubes and Rivets.

cloud of dust and smoke from under the boilers into the engine rom. After a few moments he proceeded to the stokehold and met fireman Davies trying to find his way into the engine room, and assisted him thereto. He then went back into the stokehold and assisted fireman Lynch into the engine room. On again returning to the stokehold he saw fireman Wood lying on his back apparently unconscious in front of the starboard furnace of the port boiler with part of the glowing fuel from the furnace on and around his body and his clothes partly on fire. He pulled the unfortunate man away from the fire and extinguished his burning clothing. By this time other assistance arrived, and the unconscious man was conveyed to the engine room and from thence to the saloon, where he succumbed to his injuries some three hours later without having regained consciousness. Meanwhile, the vessel had been headed towards Brest, which was about 70 miles distant, and on arrival at 3.40 P.M. on the following day the body of Wood was placed in charge of the British Consul, and both Lynch and Davies, who were seriously burnt, were sent to the hospital.

Shortly after the injured men had been got out of the stokehold, the chief engineer made an inspection of the boilers, which were then in steam communication with each other, and the steam pressure was about 125 pounds per square inch. As far as he could see or hear there was no escape of steam or water from any of the boilers. The firebars of the starboard furnace of the port boiler were all lying at the bottom of the furnace, and the smoke-box door above this furnace had been blown open. The port boiler was then disconnected from the other two and the vessel proceeded towards Brest using the center and starboard boilers only. Under these conditions the vessel was making little progress, and the chief engineer, after further satisfying himself that no part of the boiler had failed, ordered the center and port fires in the port boiler to be set away again, and shortly afterwards put it in steam communication with the other two boilers. On arrival at Brest, and while the boiler was under steam, it was again examined as far as possible by the chief engineer and a surveyor to Llyod's Register. As the latter, however, could not satisfy himself as to what had occurred he would not give a seaworthy certificate for the boiler, and accordingly the vessel left Brest for London at 6.20 A.M. on October 9 under steam from the center and starboard boilers only. With the refrigerating machinery running the available power under these conditions was considered insufficient to proceed with safety, and the vessel returned to Brest at 1 P.M. on the same day. At Brest, the port boiler was emptied, and after being roughly cleaned was examined by Mr. E. A. Thomson, Superintendent for the Blue Star Line, a Surveyor to Lloyd's Register, and the chief engineer of the vessel. As a result of that joint survey the only repair considered necessary in order to be able to steam the boiler to London was to expand six smoke tubes at their back ends in the starboard combustion chamber. This was done, and the vessel left Brest at 4 P.M. on October 11 under steam from the center and starboard boilers only, but with the assistance of a tug while steam was being raised on the port boiler. At 3.30 P.M. on the following day the assistance of the tug was dispensed with and the vessel proceeded under steam from all three boilers to London, where she arrived at midnight on October 13 without further trouble.

The Board of Trade Inspector visited the vessel on the following morning and examined the starboard furnace and combustion chamber of the port boiler as far as possible. The fires of this boiler had only been drawn a few hours previously, and there was a steam pressure of 50 pounds per square inch showing on its pressure gauge. The lower welded seam of the combustion chamber back plate was leaking slightly for a length of about 8 inches; the lower part of the combustion chamber was then practically full to the level of the furnace bridge with salt and ashes; the corruga-

tions at the bottom of the furnace were full of salt, and a lump of salt about 10 inches by 8 inches by 1 inch thick was hanging from the furnace mouth. At this time he also noticed that the portion of the bulkhead in front of the furnace mouth and at a distance of 7 feet therefrom was plastered with salt, about 9/16 inch thick, which was covered with dust. This had obviously been projected from the combustion chamber when the explosion occurred, and was, in his opinion, conclusive evidence that the explosion was caused by the rupture of a salt chamber in the bottom of the combustion chamber. Up to this time it was not known that there were any leakages in the lower portion of the combustion chamber.

On October 15 he again examined this combustion chamber and the lumps of salt deposit combined with ashes which had been dug out of it. These were from 8 inches to 10 inches in thickness and of an extremely hard rock-like structure. There was also evidence of leakage from four of the rivets in the bottom seam of the wrapper plate on the starboard side. These were subsequently carefully removed for his inspection, and it was evident from the wasted condition of their shanks, which were covered with black oxide of iron, that leakage past them had been considerable. Two days after the vessel left Cape Town the density of the water in the port boiler was 9¼ ounces to the gallon, and this gradually increased owing to leakage from the main engine condenser and the priming of the evaporator, until, on the day of the explosion, it was 14½ ounces to the gallon. Under these conditions, and with leakage from six smoke tubes, the formation of a salt chamber is not surprising.

All the essential factors for the formation of such a salt chamber were present in this case, i. e., there was considerable leakage under pressure of boiler water of high density into a space from which it could not freely escape, whereby the water, consequent upon reduction of pressure, and the temperature of the combustion chamber, was evaporated, and the salt it held in solution was deposited. This process continued and gradually filled the combustion chamber to the level of the furnace bridge with salt deposit combined with soot and ashes, forming a rock-like substance of considerable strength. A leakage into the center of this mass, such as did occur from defective rivets, would maintain a cavity or chamber containing water. The amount of evaporation from this chamber and consequently the pressure within it would depend upon the porosity of the mass, which would gradually lessen as the crown of the chamber thickened owing chiefly at this stage to the leakage from six smoke tubes, until finally the pressure in the salt chamber (which may have equalled that in the boiler) was sufficient to rupture its crown, and resulted in an explosion as previously described.

This combination of circumstances, which ended so disastrously, was brought about by the development of defects during the voyage. Similar defects, accompanied by salt deposits, are not uncommon in combustion chambers of old marine boilers; but the formation of a salt chamber containing sufficient water under pressure and of high temperature to cause an explosion of any magnitude fortunately seldom occurs. The possibility, however, of an explosion from such a cause does not appear to be sufficiently known by sea-going engineers, and it is significant that none of the engineers on this vessel had ever heard of an explosion from a salt chamber. They therefore did not suspect any danger from the accumulation of salt deposit in this combustion chamber, or it is only reasonable to assume this unfortunate explosion would not have occurred.

Since the explosion all the boilers have been surveyed by a Surveyor to Lloyd's Register of Shipping and repaired to his satisfaction.—"Marine Engineer and Motorship Builder," April, 1926.

ELIMINATING BOILER SCALE BY THE AGFIL PROCESS.

By D. A. Gardner.

One of the most important duties of the boiler plant operator is the removal of scale from boiler tubes and heating surfaces or the reduction of scale formation. Boiler incrustation is one of the serious difficulties of power-plant work and is treated in a variety of ways, depending upon such factors as feed-water composition, boiler construction and the cost of the process. The various conditions in different industrial locations—hardly any two problems are closely similar—has made the field an extensive one for engineers combating scale. Each problem must be analyzed and a remedy prescribed individually.

Scale-removal methods may be classified as mechanical, chemical or electrical. Mechanical methods attack the scale after it has formed in the boiler tubes. Chemical and electrical methods are used primarily to treat the water or the boiler surface to prevent the deposition of scale or to promote the deposition of soft scale rather than hard scale. Naturally, the latter treatments, inasmuch as they fight the scale at its source, are more desirable, but in some cases their success is limited and a combination of one of them with mechanical methods is necessary.

There are many mechanical devices on the market, and all are more or less efficacious, depending upon the hardness and thickness of the scale. Rotating weighted arms, with or without cutters, sometimes followed by wire brushes, make good tube cleaners, and a wide variety of devices have their uses in various types of boilers.

While mechanical methods prove successful in removing heat-resisting deposits, they are applied too late to be successful agents against corrosive action. The removal of solid material may destroy pockets which have been holding corrosive liquids, preventing them from being washed away by the boiler water, but mechanical treatment can seldom do much more than this. There is evidence for believing that the roughening of tubes or shells by pounding or scratching provides surfaces more susceptible to corrosion attack. Bad cases of pitting have been attributed to such roughening of tubes by mechanical cleaners.

The chemical treatment of scale comprises the addition to the water of something to react chemically with the materials already present and to convert them into substances that will not form scale under the action of heat and steam, or that will form a soft scale or mud instead of a hard, adherent scale.

For the proper application of chemical treatment it is necessary to know the chemical characteristics of the water and of the material dissolved or suspended in it, which usually requires a precise chemical analysis. In some cases activities within the boiler form materials not found in the raw water, and these do their share in the formation of scale or the corrosion of the metal, although their existence in the boiler is unknown.

Electrical methods are relatively more successful than either mechanical or chemical treatments, and are something of an innovation in the field. Unfortunately, a theoretical discussion of this method of scale elimination cannot be presented, because as yet no suitable explanation of the action has been developed.

In the past, power-plant experts have warned against the presence of electric currents in boilers, attributing corrosion and pitting to their presence. Now, however, a minute electric current passed over the inside surface of the boiler has been found to prevent the adhesion of scale to the metal and to remove any scale that may have been present before the application of current.

This process was invented in Europe and has been in use there for over three years. More than 300 successful installations are now in operation in power plants, and tests in this country have shown that the method is equally applicable to boiler conditions here. A number of installations have been in service for several months, and indications are that their performance is satisfactorily effective.

The apparatus consists simply of three pieces—a thermopile, a vibrator and an ammeter. Figure 1 is a diagrammatic sketch showing the method of connecting the outfit to the boiler. The thermopile is fastened to the steam main or to the breeching, so that one face is heated, while the other face is exposed to the relatively cool air of the room. One of the electrical terminals is connected to the upper boiler drum, the other, through a vibrator and an ammeter, to the blowoff pipe. The vibrator consists of an interrupter switch which is carried by a swinging weighted arm. The apparatus is fastened to the feed-water piping or some other vibrating part, and acts as a make-and-break shunt. This produces a direct current which pulsates more or less regularly and seems to be more effective than a continuous direct current. The ammeter serves as an indicator and also provides a means of making adjustments at the time of installation.

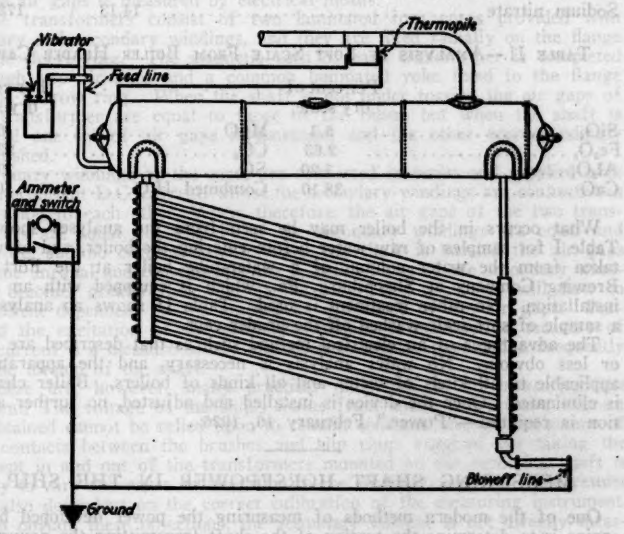


FIGURE 1.—DIAGRAM OF CONNECTIONS SHOWS SERIES CIRCUIT THROUGH THE BOILER.

When the apparatus is in operation, it seems to prevent scale-forming material adhering to the metal, and causes it to fall to the bottom. The resulting sludge is then eliminated by blowing off the boiler.

The process is very much of a mystery, and all attempts to explain it have been unsuccessful. Electrolysis, electrical repulsion and colloidal action have been suggested as probable solutions, but none has shown satis-

factory application, and in fact electrolysis has been practically eliminated because the thermopile at no time generates sufficient current to cause appreciable electrolytic action. Lack of a theoretical basis for the process is not a cause for worry, however, for it has been shown to be practicable despite its unexplainable features.

TABLE I.—ANALYSES OF RAW WATER AND OF SAMPLE DRAWN FROM THE WATER COLUMN.

	Raw-water sample.	Water-column sample.
Calcium carbonate	22.12	1.60
Free carbon dioxide	1.98	.....
Magnesium carbonate	0.77	0.63
Calcium sulphate	17.47	30.33
Magnesium sulphate	10.74	1.19
Sodium sulphate	0.98	11.08
Sodium chloride	18.15	132.00
Silica	0.88	0.72
Iron oxide	0.44	Trace
Sodium nitrate	14.13	172.09

TABLE II.—ANALYSIS OF SOFT SCALE FROM BOILER HEADER CAPS.

	Per cent of dry scale.		Per cent of dry scale.
SiO_2	5.3	MgO	6.29
Fe_2O_3	2.63	CO_2	23.80
Al_2O_3	1.20	SO_3	19.90
CaO	38.10	Combined H_2O	0.94

What occurs in the boiler may be seen from the analyses shown in Table I for samples of raw water before entering the boiler, and of water taken from the water column of a water-tube boiler at the Fort Pitt Brewing Company at Sharpsburg, Pa., which is equipped with an Agfil installation. No other treatment is used. Table II shows an analysis of a sample of soft scale washed off the header caps.

The advantages of an electrical method such as that described are more or less obvious. No water analysis is necessary, and the apparatus is applicable to all kinds of water and all kinds of boilers. Boiler cleaning is eliminated. Once the device is installed and adjusted, no further attention is required.—"Power," February 16, 1926.

MEASURING SHAFT HORSEPOWER IN THE SHIP.

One of the modern methods of measuring the power developed by an engine is to determine the torsion of the shaft transmitting the power, because that factor, in conjunction with the ascertained rigidity of the material of the shaft and with the revolutions, enables a direct calculation to be made of the power which the engine is delivering to the shaft, i. e., S.H.P.

In the case of oil engines the torque varies considerably during a revolution. A mean value of the torsion of the shaft must therefore be obtained. Several instruments are commercially produced for this purpose. One of them is here described and illustrated.

Indications of the apparatus are obtained by electrical means, the necessary current being taken from the ship's lighting circuit. With the method

employed the results are unaffected by variations that may happen to take place in the voltage of the ship's mains or be caused by dirty or indifferent electrical contacts, which lead sometimes to the failure of electrical measuring apparatus on board ship.

The value of the torsion is indicated by the reading of a rotatable divided drum against a fixed pointer, and this figure multiplied by the R.P.M. and divided by a constant gives the horsepower transmitted by the shaft at that moment.

Two split rings are clamped on the shaft with a spacing of two or three shaft diameters between them. One of the rings is narrow and the other extended in the form of a sleeve towards it, the free end of the sleeve ring being supported on the narrow ring in order to maintain it concentric. Both the narrow ring and the free end of the sleeve ring are protected with flanges.

When the shaft is under torsion the relative movement of the transverse plane through the shaft where the sleeve ring is clamped to it is transmitted by the sleeve to the plane of the narrow ring, and the flanges move relatively, in proportion to the movement between the two transverse planes through the shaft. This movement is utilized to alter air gaps in the magnetic paths of a system of differential transformers, and the alteration to the air gaps is measured by electrical means.

The transformers consist of two laminated iron cores provided with primary and secondary windings, and they are fixed radially on the flange of the sleeve ring. The magnetic path of each transformer is completed through two air gaps and a common laminated yoke fixed to the flange of the narrow ring. When the shaft is not under torsion the air gaps of one transformer are equal to those of the other, but when the shaft is twisted one set of air gaps is increased and the other correspondingly diminished.

Primary windings of the cores are connected in series and supplied with an intermittent D.C. current, whilst the secondary windings are connected in opposition to each other. When, therefore, the air gaps of the two transformers are equal, the E.M.F. of the secondary windings are equal and opposite, and no current flows. When the shaft is twisted the air gaps become unequal and consequently the E.M.F. of one of the secondary windings becomes greater than that of the other, and a current will flow to an extent dependent on the amount of difference of the air gaps. Provided the excitation of the primary windings is constant, a measure of the current is a measure of the alteration to the air gaps and consequently of the torsion of the shaft.

There are, however, many practical difficulties in so measuring the current. The voltage of the ship's mains from which the primary current is obtained cannot be relied upon to remain constant, and the resistance of the contacts between the brushes and slip rings required for taking the current in and out of the transformers mounted on the revolving shaft is likely to vary and cause an alteration to the secondary current. The results are also dependent on the correct calibration of the measuring instrument.

To obviate these objections, the secondary current is not directly measured, but is opposed by a second system of transformers exactly similar to those mounted on the shaft and in which the air gaps are alterable by hand. When the air gaps of the second or indicating transformers are so altered that an equal E.M.F. is opposed to the E.M.F. of the first or shaft transformers the alteration to the air gaps of the second system is measured and a reading proportional to the torsion of the shaft thus obtained.

In the meter the H-shaped iron piece between the cores of the transformers is moved by a micrometer screw provided with a large divided drum for indicating the movement. The "no current" or balanced condition of the two transformers is shown by a center zero indicator in the circuit.

The intermittent D.C. current for exciting the primary windings is obtained by passing the current through an interrupter actuated by a small motor, and the resultant A.C. secondary current is interrupted by the same motor to cut out one phase, so that a D.C. moving coil indicator can be used for finding the "no current" or balanced condition. To prevent rapid oscillation of the moving coil indicator electrical damping is introduced, and consequently the indication is proportional to the mean value of the fluctuating torque during each revolution of the engine.

Figure 1 shows the portion of the apparatus mounted on the shaft. The two rings can be seen clamped round the shaft, the free end of the sleeve ring supported on the narrow ring, both with flanges between which the transformers are fixed. The slip rings and brush gear for conducting the current are also shown. Parts of the protecting covers over the transformers and slip rings are removed to show the details.

The micrometer is operated by means of the knurled handle on the right hand side of the case, and the measurement is read on the drum through the window. The zero indicator is immediately above the drum and is seen through the same window. The pilot lamp, indicating when the current to the primary coils is flowing, flashes through the circular window at the top left hand corner of the case.

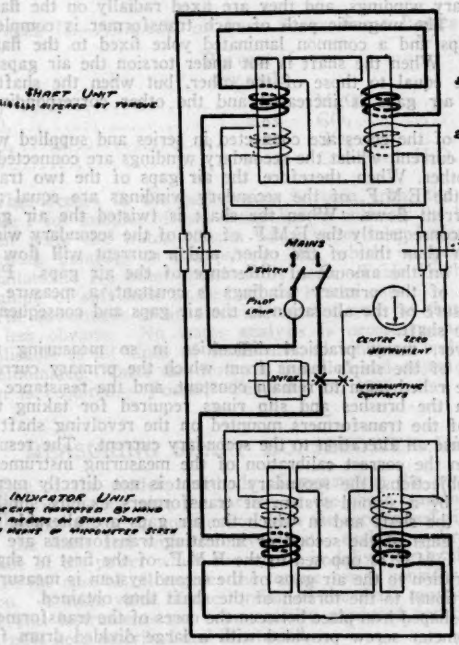

SHAFT UNIT
2 PRIMARY COILS TO GIVE SAME POLARITY
2 SECONDARY COILS IN OPPOSITION
SLIP RINGS
MAINS
PILOT LAMP
CENTRE ZERO INSTRUMENT
INTERRUPTING CONTACTS
INDICATOR UNIT
2 SECONDARY COILS IN OPPOSITION & TOGETHER IN OPPOSITION TO SECONDARY COILS OF SHAFT UNIT
2 PRIMARY COILS TO GIVE SAME POLARITY AS ON SHAFT UNIT

Diagram showing connections between coils of the shaft unit and of the indicator unit

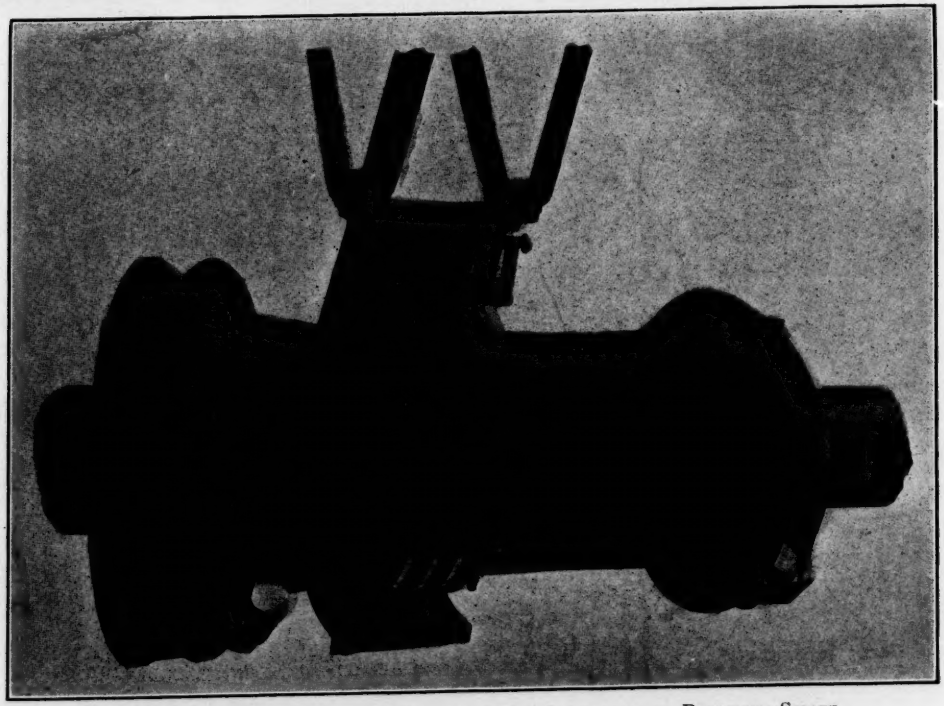


FIG. 1.—GENERAL VIEW OF THE TORSIONMETER FITTED ON THE PROPELLER SHAFT.

Means are provided in the mounting of the transformers on the shaft to enable the air gaps to be adjusted to the "no torsion" condition of the shaft after the meter is mounted.

When the apparatus is not in use the brushes are lifted from the slip rings and the current cut off by the switch lever at the right hand bottom corner of the meter.

When one desires to take readings, the brushes are first lowered on the slip rings and the current switched on. The handwheel operating the micrometer drum is then turned until the indicator pointer is at the zero position. The figure on the drum appearing against the fixed pointer is then read, and the shaft horsepower is equal to this figure multiplied by the R.P.M. of the shaft and divided by a constant. The constant is obtained either by a direct calibration of the shaft and torsion meter together or by calculation from the known dimensions of the shaft, the ascertained or assumed modulus of rigidity of the shaft and the known characteristics of the meter.

This particular type of torsionmeter is manufactured by Siemens Bros., Ltd., of London, and is supplied in this country by the McNab Co., of Bridgeport, Conn.—"Motorship," April, 1926.

THE PHOTO-ELECTRIC CELL AS A SMOKE DETECTOR.

S. H. Reynolds.

The rise of radio to the heights that we know it today, began with the perfection of the vacuum tube. The entire radio industry would soon be a memory should the tiny electrons which fly off from the hot filament of the vacuum tubes go on a strike and refuse to act in the manner we have learned to expect.

Another electron device has made its appearance in late years and is beginning to find uses in radio as well as in various other industries. It, also, is a vacuum tube, constructed and operated quite differently from the familiar radio tube, in that its action is controlled by light. This device is known as a photo-electric cell, and it has been taking a more and more important place in laboratories, as a precise and sensitive apparatus.

Certain metals have the property of giving off electrons when their surface is illuminated, and this is the basic principle on which the photo-cell operates. Under ordinary conditions, with the metal's surface exposed to the air, the electronic emission is imperceptible, and the electrons are lost in the immense swarm of air atoms which surround the metal. If, however, the metal is put in a vacuum and a beam of light thrown on it the electrons are not stopped by the air atoms. The showers of electrons then become quite perceptible. If a collector with a suitable positive potential impressed on it is put in the vacuum with the illuminated metal plate or metallic substance, the electrons will be attracted to it and will give a quite measurable current, just as the electrons do which are driven off by the heat from the filament of the radio tube. So long as the beam of light continues to fall on the metallic substance the electrons will be emitted and the current due to them will continue to flow.

Methods have been developed for the amplification of photo-electric impulses by means of thermionic tubes, and after this development the application of cells extended far beyond the laboratories. The photo-electric impulses can be amplified as much as a million times by means of a single thermionic tube. These methods are particularly adaptable for amplification of very weak impulses, and require careful insulation and even special thermionic tubes for best results.

In order to simplify the installation and adapt the photo-electric cell for the use of untrained operators, a device has been developed by Mr. V. K.

Zworykin to meet these conditions, having in view a power output sufficient to operate directly the average mechanical relays. This device, as shown in Figure 1, is a combination of what is practically a standard three-electrode thermionic tube with a photo-sensitive control electrode. The lower part, resembling an ordinary radio tube, consists of an oxide coated filament which operates at a temperature below visual emission, and the usual grid and plate, all mounted on a standard base. One-half of the inside of the upper part of the cell is coated with a substance, such as an alkali metal, which emits electrons under the influence of light. To prevent any possibility of light from the filament falling on the sensitive coating, the upper part of the cell is shielded from the amplifying portion of the tube by metal shields, which serve also to connect the grid to the coated surface of the upper part of the cell. The electron collector, shown as a loop in the upper part of Figure 1, is connected to the plate of the amplifying element. Great care is taken to prevent the alkali metal coating from condensing appreciably on the insulating parts of the cell. The connections for this cell when used as a smoke detector are shown in Figure 2. When in the dark, the action of the upper part of the cell is negligible, and the lower part acts as an ordinary three-electrode tube. The very high grid bias, or negative charge impressed on the grid by the C battery, completely neutralizes the attraction of the positive charge on the more distant plate for the electrons given off by the filament, so that the current in the

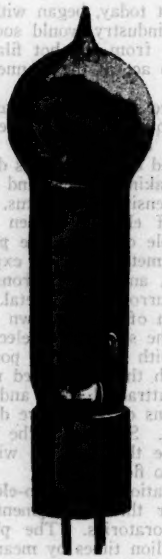


FIG. 1

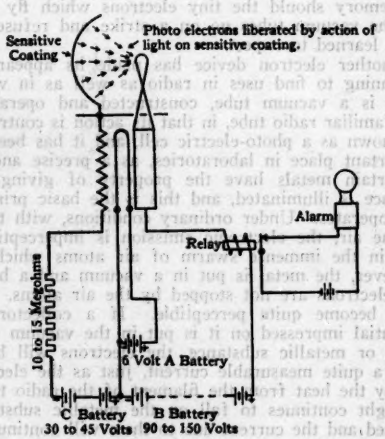


FIG. 2

FIG. 1—THE PHOTO-ELECTRIC CELL

FIG. 2—CONNECTION DIAGRAM OF PHOTO-ELECTRIC CELL USED AS SMOKE DETECTOR

plate or B battery circuit is practically zero. So long as no current is flowing in the grid circuit, the effect of the high resistance in this circuit is also negligible.

If, however, light falls upon the sensitive coating, electrons are given off and these are attracted to the collector by its positive charge, which equals the sum of the B and C battery potentials. The resultant current is proportional to the brightness of the light which falls on the sensitive coating, being independent of the applied voltage, provided this is sufficiently great to attract to the collector all the electrons that are given off. This current in the light sensitive part of the cell is very small, being only a fraction of a micro-ampère. It does not have any appreciable effect on the relay. It is, however, sufficient to produce a voltage drop over the high resistance in the grid circuit and to partly or even completely neutralize the negative grid bias on the three-electrode amplifier, permitting a flow of current in the amplifier plate circuit which is many times that in the upper part of the cell. This current is still quite small (several milliampères), but is sufficient to actuate the relay. Within limits, this amplifier plate current is proportional to the amount of light which falls on the light sensitive element of the upper cell. There is, of course, a certain time lag between the fall of light on the sensitive coating and the resultant flow of current in the plate circuit of the amplifier. This time lag has been proved by test to be less than 1/10,000 of a second and is estimated to be of the order of 1/100,000 of a second.

When used as a smoke detector this cell is located in the end of a tube to prevent light from an outside source affecting it. A beam of light is passed over the apparatus to be protected and falls on the sensitive coating, thus maintaining current in the relay circuit and hence keeping the alarm circuit open. The alarm circuit, such as a bell, may be of any desired type which can be operated from a relay. Any smoke arising from the apparatus will diminish the intensity of the beam of light, causing the relay to actuate the alarm. The relay and the photo-electric cell circuits can be adjusted so that a slight diminution in the light will actuate the relay and give a signal.

This application is one of the many to which the photo-electric cell is suitable. It is being used for the telegraphic transmission of photographs and may before long be used for transmission of vision by wire or radio. It can be used by the motion picture industry to determine the relative exposure of negatives taken under different conditions. It may be used for determining the density of liquids, to sort cigars by color, or determine the difference between slight shades of cloth, or as a burglar alarm, and in many other ways.—"Electric Journal," March, 1926.

ARC WELDING IN HYDROGEN AND OTHER GASES.

By P. Alexander,

Thomson Research Laboratory, General Electric Company.

The growing importance of electric welding in manufacturing and structural work is constantly urging research engineers to pursue their work in improvements of methods and equipment with increasing energy. It is the purpose of this article to give a brief description of a new method of arc welding in a hydrogenated atmosphere.

The two problems that have attracted most attention in the last few years are greater speed of welding and improvement in ductility in the deposited metal. The first problem up to a certain limit can be solved by a continual increase in the welding current. The second is more difficult

and is still in process of actual development. The reason for this is the exceptionally high temperature of the arc and the very short time in which chemical reactions and solidification of the metal are taking place.

The extensive research work carried out during the last decade by several laboratories and a number of isolated investigators has shown that the ductility of the weld is limited by the presence of oxides and nitrides of iron introduced into the molten metal during the welding operation.

Aiming first at the solution of the second problem; that is, the production of more ductile welds, the author has been conducting a systematic investigation on welds produced in different gaseous media.

Since the accepted explanation of brittleness in arc welds was the presence in the weld of oxides and nitrides of iron, the first experiments were conducted with the idea of complete elimination of both oxygen and nitrogen in the atmospheric gases around the arc.

THE HYDROGEN WELD.

After a series of preliminary experiments with carbon dioxide, superheated steam, illuminating gas, and other gaseous mixtures, pure hydrogen was tried, which at once gave excellent results. The welds produced in that gas proved to be perfectly ductile, to be almost free from oxide inclusions, and to possess a very high tensile strength.

The practical solution of supplying a hydrogen atmosphere around the arc consists in directing a jet of hydrogen alongside the welding electrode. In other words, the arc is drawn inside of a hydrogen flame.

This method at first met with great difficulties on account of the instability of the arc in that gas. But subsequently that obstacle was not only overcome but turned into a great advantage.

The first condition for drawing an iron welding arc in hydrogen is an open-circuit voltage of the generator of at least 120 volts. The second characteristic of the welding arc in hydrogen is a high voltage drop across the arc, which for practical purposes is always about forty volts.

The sum of the anode and cathode drops in the welding arc of 100 ampères was found to be about 30 volts, which is more than double that of the same arc in air.

From this it follows that with the same welding current the apparent double resistance of the arc permits the liberation in the arc of double amounts of energy.

This advantage has been utilized still further by using arcs of 55-60 volts. In that case the energy of the arc is trebled and is sufficient to do welding of the stock up to half an inch in thickness without beveling the edges of the plates. This in itself is a considerable saving in time and cost.

The average speed of welding attained with the apparatus shown in Figure 1 on ¼-inch and ½-inch plates was considerably higher than that attained in air even with slightly higher currents.

SPEED.

The high speed of the hydrogen weld is due to two causes—first a much greater energy of the arc and second to the welding without beveling of the edges. In such case the amount of the metal which it is necessary to deposit to fill up the gap between the plates is very small indeed. The operation of joining two plates edgewise is reduced to a rapid melting of the edges together and a compensation for the metal lost by an electro-evaporation from the crater.

To give a specific example of the speed of welding, it can be stated that by using 180 ampères and 60 volts, ¼-inch boiler plates have been welded at a speed of 60 feet per hour.

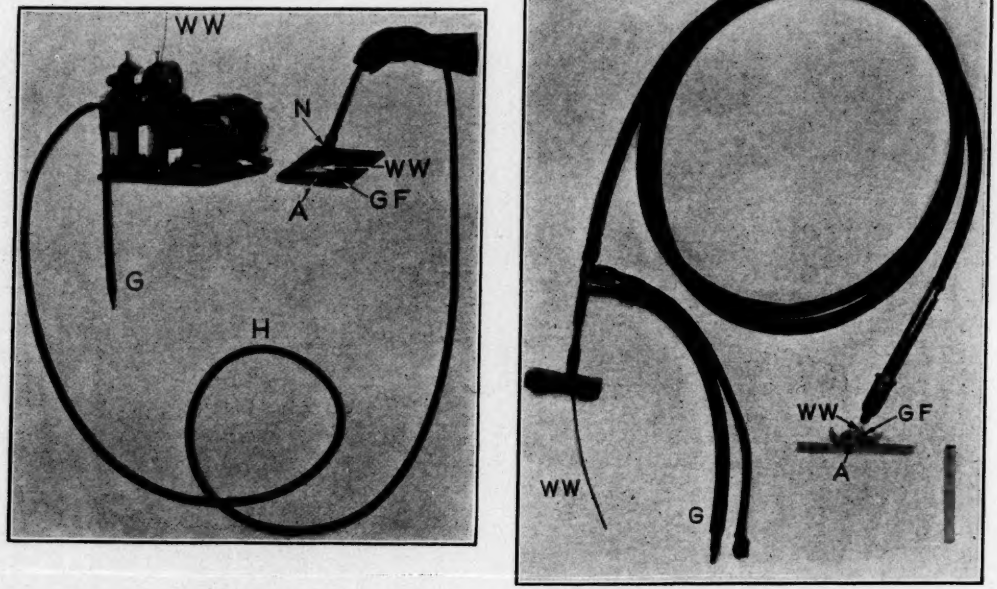


FIGURE 1.—APPARATUS USED IN THOMSON RESEARCH LABORATORY FOR WELDING IN A HYDROGENATED ATMOSPHERE WITH METALLIC ELECTRODE.

A—Arc
G—Gas Hose
GF—Gas Flame
H—15-foot Flexible Hose Conducting Gas, Electric Current, and Welding Wire
N—Nozzle
WW—Welding Wire

The second feature of the process is even more valuable than the speed, in that it gives a weld of great ductility.

Data on test bars machined out of metal deposited entirely by the arc presented in Table I, an inspection of which will give an idea of the physical properties of the deposited metal.

Material	Electrode Used	Carbon Content of the Deposited Metal	Elastic Limit	Tensile Strength	Reduction of Area Per Cent	Elongation Per Cent
Arc Deposited Metal	G. E. Sheathed Electrode	0.05	42,000	57,800	55	36
			39,375	55,435	49	31
			42,685	56,750	40	29.5

Particular attention is called to its remarkable ductility. This is a factor of prime importance in the welds of structures that are subjected to vibration, accidental bending stress, or overload. Also, it equalizes the internal cooling stresses when present in the weld.

If any part of the ductile weld is stressed beyond its elastic limit, it will not crack. It will yield until the stresses are more or less equalized all along the joint, which is so proportioned as to stand with safety the imposed load.

As it was stated, the arc is maintained inside of a hydrogen stream which burns along its outer surface of contact with the air. The electrode is entirely surrounded by hydrogen, which eliminates the possibility of the metal in the crater coming in contact with air.

At high temperature hydrogen acts not only as a blanketing medium but also as one of the most energetic chemical reagents. Besides this action of molecular hydrogen a certain amount of the gas in the vicinity of the arc core is dissociated into the atomic state and its exceptional affinity for oxygen is sufficient to prevent any oxidation of the weld.

Neither the oxygen produced by the decomposition of oxides present in the metal plate nor that which infilters through the hydrogen flame will combine with iron so long as a sufficient amount of hydrogen is present.

The active nature of hydrogen is of paramount importance for practical applications of the process. As practically all arc welding is done in open air, it is impracticable to devise a torch which will absolutely exclude all the air from around the arc, and unless hydrogen neutralizes the action of infiltrated oxygen and nitrogen the results cannot be uniform.

Summarizing the above brief description of the hydrogen process, it can be stated that it accomplishes two main purposes; first, it increases the speed of welding by concentrating in the arc large amounts of energy without the use of excessive currents; and second, it produces a perfectly ductile weld.

The continuous absorption and evolution of large amounts of hydrogen by the molten metal is equivalent to a thorough washing of the liquid metal with hot hydrogen. It is believed that the exceptionally high elastic limit of the deposited metal is due to this washing with atomic hydrogen. Indeed the elastic limit of the pure iron, electrode before deposition is on an average 29,000 pounds per square inch. The elastic limit of the same electrode deposited by the arc in hydrogen is on an average 42,000 pounds per square inch.

WELDING IN WATER GAS ATMOSPHERE.

After the work on the hydrogen welding had been found to be successful as stated, Professor Elihu Thomson suggested the great desirability of testing whether mixtures of gases containing hydrogen or substances which under the heat of the arc would produce hydrogen along with carbon monoxide would not give a thoroughly practical result, such as would meet the necessities of the ordinary art. One of the first of these tried was actually water gas, which is a mixture of equal volumes of carbon monoxide and hydrogen gas. It was established that a weld produced in water gas or certain other gaseous mixtures or mixtures of gases and vapors is perfectly ductile and easier to produce than when made in substantially pure hydrogen. It was found that the open circuit voltage of the welding generator need not be abnormally high and that the arc itself became much more stable. Other investigations were carried on with various mixtures in which the substantial composition of the resulting gases, when exposed to the heat of the arc, would be of the same nature as that of water gas. Another advantage of water gas is its exceedingly low price.

The usual method of generation of water gas is by passing a jet of steam over the surface of incandescent coke. This operation is conveniently carried out in a special gas generator consisting of a furnace filled with slowly burning coke, and a boiler. The steam generated in the boiler is passed through the pile of incandescent coke. At this temperature and in the presence of carbon, water or steam is decomposed. The oxygen combines with coke, producing carbon monoxide, which after mixing with liberated hydrogen gives a gaseous mixture of 50 per cent hydrogen and 50 per cent carbon monoxide by volume, known as water gas.

The work with various mixtures of carbon monoxide and hydrogen, produced either synthetically or by decomposition of various organic compounds, demonstrated that ductile welds can be produced in the atmosphere of any mixture of hydrogen and carbon monoxide.

The mixture of carbon monoxide and hydrogen can be produced in special generators or, as in case of a mixture of propane and carbon dioxide, immediately in the flame surrounding the arc.

Among all the gases and organic compounds, methanol deserves a special consideration. Methanol or synthetic wood alcohol, designated by a chemical formula CH_3OH, is a liquid at ordinarily temperature. It boils at 66 degrees C., and the vapors are completely decomposed into hydrogen and carbon monoxide at about 700 degrees C.

This shows that a gaseous mixture of a similar composition to water gas can be stored in a perfectly stable liquid form. It can be easily transported with the portable welding outfits and, when desired, readily converted into a gaseous mixture. The significance of the storage of gases in a liquid state can be seen from the fact that one gallon of methanol will give 232 cubic feet of gas.

Another great advantage of methanol is that carbon monoxide and hydrogen are stored in a chemically combined state, which eliminates any danger from poisonous action of the former gas, in case of a slight leakage from the storage container.

The decomposition of methanol vapor into a mixture of hydrogen and carbon monoxide can be effected in a special dissociation chamber or inside of the flame played from the nozzle of the welding torch. At the surface of the flame these gases combine with the oxygen of the air and, due to the excess of that gas, are completely converted into water vapor and carbon dioxide. The welding torches with metallic or carbon electrodes are the same as those used for welding in pure hydrogen.

Methanol is a cheap commercial product and the latest developments of producing it on a large scale indicate that in the very near future the welding in dissociated methanol will be almost as cheap as in water gas.

WELDING IN NITROGEN-HYDROGEN MIXTURES.

The idea of welding in an atmosphere of neutral gases is not a new one. Carbon dioxide, which is an example, although neutral at ordinary temperatures, is readily decomposed by the high temperature of the arc into a mixture of oxygen and carbon monoxide, and therefore the weld produced in a CO_2 atmosphere is highly oxidized and is quite brittle.

Welding in nitrogen has been tried by many investigators, and the results obtained were not at all encouraging. Lately the extensive work on nitrides in steel and electric welds demonstrated the deleterious effect of nitrogen at high temperatures on ferrous alloys.

These various investigations demonstrated without room for doubt that if the nitrogen content in steel reaches 0.05 of one per cent the metal is so brittle that it cannot be used in engineering structures. Mixtures of nitrogen with carbon monoxide have also been tried without success. It seemed that nitrogen was barred from any welding work on ferrous alloys. Yet Dr. Irving Langmuir and the author, after examination of all the evidence investigated once more the possibility of using this gas for welding, to ascertain whether some of the mixture of this gas with various amounts of hydrogen could not be used for welding work.

A series of experiments conducted with the nitrogen-hydrogen mixtures demonstrated that mixtures of these gases give perfectly ductile welds and that the amount of hydrogen might be varied within very wide limits. It might be as high as 90 and as low as 10 per cent. If the proportion of either gas exceeds 90 per cent the stability of the arc and the quality of the weld are characteristic of the predominating gas. By taking special precautions and by using gases of very high purity, the percentage of hydrogen may be reduced to 6 per cent or even less, and still produce fairly ductile welds. Yet for practical purposes when the welding is done under factory conditions, 8 per cent of hydrogen should be regarded as the lower limit. The presence of nitrogen in the gaseous mixture is manifested by a much lower striking voltage and a greater stability of the arc. The arc voltage also is greatly influenced and, as the experiments indicated, is directly proportional to the amount of hydrogen. The possibility of using hydrogen-nitrogen mixtures enabled another solution to the problem of storage of welding gases to be worked out.

It must be remembered that pure hydrogen cannot be dissolved to any considerable extent in any common substance. The only way to store hydrogen is by compressing it into high-pressure cylinders. But even compression as high as 2000 pounds per square inch permits the storage of only 200 cubic feet of gas in a cylinder of the usual size.

To store the gas in large quantities it must be liquefied. Following this line of investigation the author conceived the idea of using ammonia. This substance is represented by the chemical formula NH_3, which indicates that if this compound be dissociated into its elements, it will give a stable gaseous mixture consisting by volume of 75 per cent of hydrogen and 25 per cent of nitrogen.

The following work with this substance demonstrated that the decomposition can be accomplished very conveniently in the presence of the usual catalysts at a temperature as low as 600 degrees C.

The apparatus used for welding in dissociated ammonia consists of the ordinary storage cylinder containing 50 or 100 pounds anhydrous liquefied ammonia and a dissociation chamber about 12 inches long provided with the catalyst and a heating coil. Once ammonia has passed through the dissociation chamber, it is entirely decomposed into a perfectly stable mixture of hydrogen and nitrogen which may be used immediately in the welding torch or stored in a gasometer. This mixture can be used in the same way as pure hydrogen for welding with metallic or carbon electrodes.

The significance of using ammonia can be seen from the fact that a comparatively small cylinder of 50-pound capacity will supply 2250 cubic feet of gas. The larger cylinders of the size of low-pressure hydrogen cylinders used in the factories contain 100 pounds of liquid ammonia and will produce 4500 cubic feet of gas.

The present cost of liquid ammonia is fifteen cents per pound, so that the cost of the gaseous mixture will be one-third of one cent per cubic feet, which compares very favorably with the cost of hydrogen sold in high pressure tanks at one cent per cubic feet.

CONCLUSION.

In the outlined methods of welding and the obtaining of protective gaseous media for the operation, emphasis was laid on hydrogen, water gas, methanol and ammonia, since these gases are likely to find immediate industrial application on a large scale.

The welding in other media and mixtures has been also thoroughly investigated. The results of those experiments are of a somewhat more academic nature and will be published in a separate article devoted to the theoretical aspect of the welding in different gaseous media and the mechanism of nitrogenation of the arc-fused metal.—"General Electric Review," March, 1926.

TRANSATLANTIC WIRELESS TELEPHONY.

At the beginning of 1923 we had the privilege of being present at a demonstration of wireless telephony across the Atlantic. At the New Southgate works of the Western Electric Company—now Standard Telephones and Cables, Ltd.—we listened, during the early hours of January 15th of that year, to speech transmitted from 195 Broadway, New York, and, as reported in our issue of January 19th of 1923, the messages which were received over a period of practically two hours were replied to on this side by cable. The demonstration indicated that sooner or later a commercial system of wireless communication might be established between the two countries, although the engineers in charge of the test made it clear that a good deal more experimental work would be necessary before it could be definitely decided whether such a scheme would be practicable.

In all electrical communication systems, irrespective of whether wires are used or not, noise tends to make the signal unintelligible, and if the noise measured in the same units as the signal is nearly equal to or is greater than the signal, no amount of amplification will make the signal distinct. For good and reliable communication the strength of the received signal must be considerably greater than that of the noise, the limit of transmission range being set by the allowable disturbance ratio. The problems connected with simultaneous two-way wireless telephone transmissions are naturally much more difficult than those of one-way transmissions, particularly where land telephone lines are also involved, and many difficulties, including those arising from interference, have to be surmounted. Although the daily newspapers have forecasted the early inauguration of a Transatlantic radio service and have actually announced the probable charges which will be made for calls between this country and America, in official circles these statements are considered to be altogether too premature.

After the Transatlantic test that was carried out in 1923 a committee was appointed by the Postmaster-General, under the chairmanship of Admiral Sir Henry Jackson, to investigate the possibility of a commercial wireless service, and during the past two years further experiments have been conducted under the supervision of this committee and in co-operation with the British Post Office. As our readers are well aware, wireless

telephony transmitting apparatus has been installed by the Post Office in the new wireless telegraphy station at Rugby, thus making two-way experimental communication possible. During the course of ordinary routine work on February 7th two-way conversation tests were made from 4 A.M. to 7:30 P.M. with very promising results, the tests being undertaken by the station staffs at Rugby and in America. At the time the conditions of the ether were abnormally favorable for wireless transmission, for there was little atmospheric disturbance or noise. The quality of the speech in both directions was clear and free from distortion or fading, and the voices of the individual speakers could easily be recognized.

On Sunday, March 7th, another very successful two-way test was made, when we had an opportunity of carrying on a two minutes' conversation with one of the engineers at the headquarters of the American Telegraph and Telephone Company at 24 Walker street, New York City. For the first time in the history of wireless telephony a group of persons, consisting largely of representatives of the daily Press, was able to listen to conversation across the Atlantic and individually to converse with someone in America. Incidentally the demonstration took place on the fiftieth anniversary of the telephone. In England the circuit terminated at the London trunk exchange of the British Post Office, whilst in America it ended at the long-distance telephone headquarters of the American Telegraph and Telephone Company. As the diagram Figure 1 indicates, in transmitting from London the speech is carried by underground telephone wires about 80 miles long to the Post Office Rugby radio sending station, and thence by wireless to a radio receiving station at Houlton, Maine, U. S. A., a distance of 2900 miles, whilst from Houlton the speech is passed over 600 miles of telephone circuit to New York City. In transmitting from America the speech is carried from New York to the sending station of the Radio Corporation at Rocky Point, Long Island, a distance of 70 miles, and from there it is sent by wireless over 3300 miles to a receiving station at Wroughton, near Swindon, where there is a Beverage aerial about 7 miles long. It is well known that this type of aerial has very marked selective properties and that it consists of a long and low horizontal wire pointing in the direction of the transmitting station. From Wroughton the speech is carried over a 90-mile telephone circuit to London. The wave length utilized in England is 5770 meters, whilst that employed in America is 5260 meters.

On Sunday last the reception in this country left little, if anything, to be desired, and judging by the ready manner in which those at the other end of the system answered questions, it was equally good in America. Everyone present at the trunk exchange in Queen Victoria street was provided with a pair of headphones and all had an opportunity of speaking to America, whilst the others listened to the conversation in both directions. The only hitch in the whole business was that a number of American journalists, who should have been at Walker street, New York, at 9 A.M. (American time), to converse with representatives of various journals in this country, did not arrive until considerably later, with the result that some American engineers had to take their places.

Great credit is due to the engineers of the wireless section of the British Post Office for the manner in which the demonstration was organized. Before the test commenced Mr. E. H. Shaughnessy, who is at the head of the Wireless Department, delivered a brief and popular lecture on the complete system, and by means of lantern slides gave some idea of what wireless telephony across the Atlantic really entails.

Technical details of the wireless telephony apparatus at Rugby are apparently to be given in a forthcoming paper by Mr. E. H. Shaughnessy, who in the past has been kind enough to permit us to publish a good deal about the telegraphy section of that station. For the present, therefore, we cannot

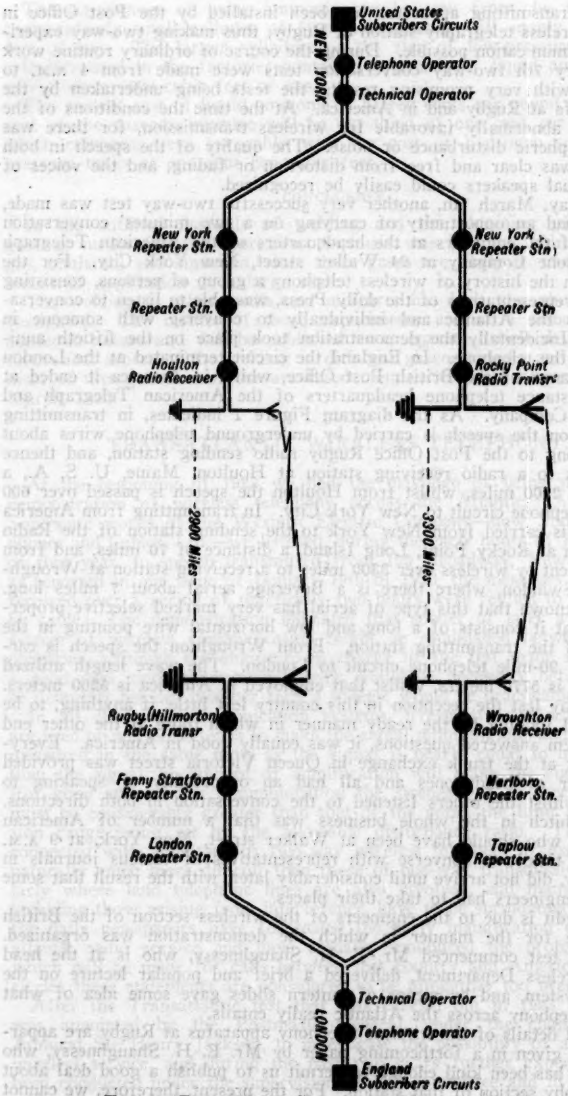


FIG. 1.—DIAGRAM OF CABLE AND WIRELESS CIRCUITS.

do more than refer in a general way to the system of radio communication that has been adopted, namely, the so-called single-side band system. Before the late Dr. Nichols, who was in charge of the test made in 1923, returned to America shortly after the test, he delivered a lecture before the Institution of Electrical Engineers on this particular system, which offers certain advantages, and in view of the renewed interest that has been aroused in the matter by last Sunday's test, we propose to give a brief summary of that lecture. In order to introduce the subject as easily as possible, he first considered the simple type of circuit shown in Diagram I, Figure 2, where *a* represents a telephone line, *b* a battery which sends direct current through the circuit, and *c* the microphone, which varies the amplitude of the current or "modulates" the carrier current. The telephone receiver *d* consists of a diaphragm on which the electro-magnet can operate. When the microphone is operating, the amplitude of the carrier current, which is otherwise constant, is slightly varied, with the result that the current which flows through the telephone receiver is made up of two components—the constant carrier current or something almost exactly equal to it and the signal in the form of a variable ripple superimposed on the carrier. In the telephone receiver a deflection of the diaphragm is produced which is proportional to the square of the current passing through it. This is indicated by

$$(Ic + Is)^2 = Ic^2 + 2Ic\,Is + Is^2.$$

If Ic represents the constant carrier current and Is the signal the force on the diaphragm of the receiver is proportional to the square of $Ic + Is$, which is made up of three terms, Ic^2, which is constant, the middle term $2\,Ic\,Is$, which is directly proportional to the signal, and Is^2, which is proportional to the square of the signal, and consequently of double frequency, because any periodic function squared gives a constant and a double frequency component. The third term is consequently not required. If the received speech is to be of good quality the term Is^2 must be small as compared with the $2\,Is\,Ic$ term—*i.e.*, the carrier must be rather large as compared with the signal, and this condition is not met in the case of the simple telephone circuit No. 1 in Figure 2, which possesses two or three disadvantages applicable to a wireless circuit. The battery current or "carrier" current which flows through the line contains no element of the signal, and there is consequently really no need to transmit this direct-current carrier from the transmitting end to the receiving end, except as a means of producing the middle term $2\,Ic\,Is$ at the receiver. Consequently the transmission of the carrier current through the line involves an unnecessary loss of power. Moreover, if the characteristics of the line vary, both Ic and Is in the middle term will vary and the variation factor in the received signal will be squared.

An obvious improvement can be made by employing the arrangement No. II. in Figure 2, where transformers are introduced at the transmitting and receiving ends of the line. Clearly the first transformer restricts the carrier current to the transmitting apparatus, and the variations in this carrier current go through the transformer to the receiving end, where only double frequency signals would be obtained, because the only thing that appears in the receiver is the Is term. To make the signals intelligible it is necessary to re-introduce the carrier current Ic, which is done either by placing a battery at *e* or a permanent magnet in the telephone receiver, thereby restoring the conditions which existed at the receiving end under the conditions indicated by the diagram No. I. In the diagram No. III., Figure 2, the distribution of currents which must be passed by the line is plotted as a spectrum. In order to transmit intelligible speech a range of frequencies of something like 3000 periods is required, and the microphone generates and the line must transmit a spectrum represented roughly by the rectangle on the left of diagram No. III. In diagram No. IV., Figure 2,

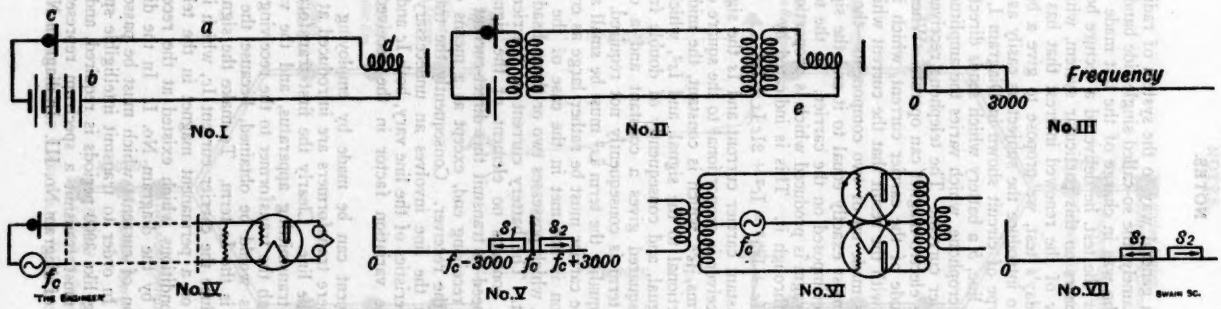


FIG. 2.—DIAGRAMS EXPLAINING THE PRINCIPLE OF SINGLE SIDE BAND WIRELESS TELEPHONY.

exactly the same arrangement is shown with the exception that the carrier current is an alternating one of frequency fc and is delivered by an alternator. The microphone modulates the alternating carrier current and the result is to produce a spectrum represented by s_1, fc, and s_2, as in the diagram No. V. There is a carrier frequency component fc and a band called the upper side band extending up to the carrier frequency plus 3000 periods, and also another band called the lower side band, extending down to the carrier frequency minus 3000 periods; in other words, the effect of modulating an alternating carrier by speech is to produce a spectrum twice as wide as the band which it is necessary to transmit. At the receiving end there is a device which operates in a similar manner to the telephone receiver; that is to say, currents in the output circuit of the valve are proportional to the square of the voltage applied across its input terminals. This means that the valve conditions are so arranged that if a sine wave be applied to the grid circuit a double frequency will be obtained in the output circuit. Exactly the same remarks apply in this case as in the case of the simple telephone circuit. A current is transmitted over the line having a spectrum as shown by the diagram No. V. The carrier component has no element of signal in it and there is no need to transmit it over the line. If it is transmitted, then the effects of variation in the line will be squared in the receiving circuit, and, moreover, it will be necessary to use much more power than would be required if it were not transmitted. Two important advantages are therefore to be derived from the practice of excluding the carrier wave from the line itself.

The carrier current may be excluded by means of the device shown in diagram No. VI, although other arrangements may also be employed. The carrier current is generated by means of the alternator shown at fc, and the speech which modulates this current is applied at the left. If no speech is applied and if the valves and transformers are quite symmetrical, the carrier current will produce no effect whatever on the output circuit, because currents will flow down in the upper part of the transformer winding and equal currents will flow up in the lower part of the winding. As soon as the microphone is spoken into, however, the valves will be unbalanced and the output will be proportional to that unbalance, the peculiarity about the output being that there will be no carrier frequency component present, as indicated in the diagram No. VI. Instead of obtaining the original spectrum as indicated in diagram V., a new one will be obtained which contains only one-third of the power of the original spectrum, but which still contains all the elements of the signal. As the two side bands are exactly alike, it is not even necessary to transmit both of them. One of them—say, the upper band—can be cut out by means of electrical filters and all the elements of the signal will still be retained. If, however, at the receiving end the side band be applied directly to the receiving valve, the signal obtained in the output circuit of that valve will be unintelligible, because it is proportional to the square of the input and it will be of double frequency. If the carrier component be introduced at the receiving station in the same way as that described in connection with the simple telephone circuit and the signal then be detected, exactly the same condition will be secured as before. A component will be obtained that is directly proportional to the signal, which is what is required.

Wireless transmission must, of course, take place through what is known as the ether, which is more like a condenser than a copper wire, and consequently in order to get signals through it is necessary to use rather high frequencies. Moreover, the transmission characteristics of the ether are very much more variable than those of a wire. By employing the single-side band system there is a considerable saving in power, and the minimum frequency range is utilized in the ether, thus making more channels of commutation available.

To what extent the apparatus used for the recent tests differs from that employed for the American transmission in 1923 we cannot say, but presumably it is fundamentally the same. We commence, said the late Dr. Nichols, in his lecture before the Institution of Electrical Engineers, by developing a modulated wave at very low power by means of valves as used in telephonic repeaters. Both side bands are generated, but not the carrier, which is eliminated by the balanced modulator system shown in diagram No. VI. in Figure 2. By means of filters one of the side bands is cut out, because both bands are unnecessary, and a single band high frequency telephone current is produced. The next step is to amplify the low output up to about 300 watts and then, by means of another amplifier, up to about 5 kilowatts, the final stage of amplification taking place in a still larger amplifier capable of raising the power up to 200 kilowatts, which supplies the aerial. Only one side band of the modulated wave travels across the ocean, and, as explained above, by analogy, if an attempt were made to detect it alone, the speech received would be unintelligible, because all the frequencies would be doubled. Thus before detection is undertaken the carrier frequency component must be re-introduced—that is to say, the first frequency is generated as at the transmitting end. An oscillator is made to generate a current of this frequency and it is imposed on the receiving circuit and finally the combination is detected when the original speech is obtained, the resultant output of the receiver being amplified as much as may be necessary.—"The Engineer," March 12, 1926.

THE WIRELESS EQUIPMENT OF A SEAPLANE.

Wireless apparatus now forms an important part of the equipment of aircraft used for combatant services or for commercial purposes, and not the least of the many factors contributing to the successful Transatlantic flight of Commandante Franco was the use of the wireless apparatus installed on the Dornier seaplane, the *Ne Plus Ultra*.

The Dornier Wal seaplane is constructed of metal and was equipped with a Marconi standard type A.D. 6 telegraph-telephone set and a Marconi direction-finding set.

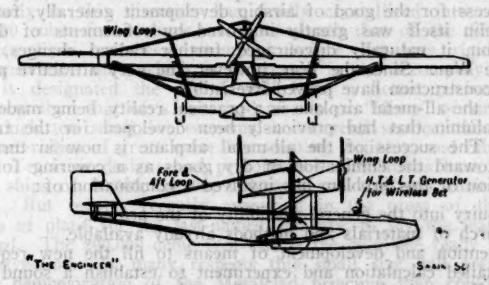
A special type of loop aerial was fitted on the machine, and the Marconi standard aircraft direction-finding receiver was altered to increase the working range of the equipment.

The fore and aft loops consist of two single-turn loops supported on struts and arranged symmetrically on each side of the hull, great care being taken to avoid any out-of-balance effects owing to the presence of large masses of metal. The two loops are connected in series, and the total area turns is approximately 250 square feet. In order to obtain equal reception with the wing coil, a single turn loop is mounted in the plane of the wing. The two sketches below show the positions of the two fore and aft loops and the wing loop. The terminal ends of the loops are connected by a length of cable leading to the navigator's position near the pilot.

To increase the working range of the direction finder and to compensate for the smaller receptive power of the loops, it was found necessary to increase the overall magnification of the standard aircraft direction-finder receiver. This was accomplished by the use of a twelve-valve amplifier divided into two units, the first unit containing six stages of high-frequency amplification and a detector valve, and the second unit an oscillator valve, two stages of intermediate frequency magnification, a detector valve and a note-magnifying valve. The anode voltage for the receiver is supplied

by a ninety-six-volt dry cell battery and the filaments are run off an accumulator battery. The radiogoniometer and high-frequency tuning unit are of the standard type supplied with aircraft equipments, and cover a wave range of 600 meters to 1000 meters. The equipment was calibrated on wave lengths of 600 meters and 900 meters, and the quadrantal error curves were well within the limits required for navigational purposes.

Reports from Captain Ruis de Alda, the navigator who took the direction-finding bearings during the flight, show that bearings were obtained at a distance of 700 kilometers from Las Palmas, and such was the accuracy of the readings that the course was directed solely by the aid of the direction finder during this stage of the flight. When nearing Las Palmas visibility was very poor, but the signals from the shore stations enabled the navigator to maintain a direct course to his destination.



FLYING BOAT FITTED WITH WIRELESS EQUIPMENT

The standard type A.D. 6 telegraph-telephone set was used for the transmission and reception of ordinary service messages, the essential components of both transmitter and receiver being mounted in one unit. The transmitting system comprises an aerial tuning inductance, an oscillation valve, control valve, and a sub-control valve; whilst the receiving system comprises an aerial tuning inductance and a five-valve amplifying detector.

On the transmitter two Marconi standard type M.T. 3 F. and one M.T. 5 are used, the total filament current being 6.6 ampères at a pressure of 6 volts. For reception two type V. 24 and three type Q X valves are employed, with a total filament current of 3.75 ampères at a pressure of 5 volts. The wave range of the transmitter and receiver is variable between 400 meters and 1200 meters. The system of remote control by means of Bowden cable enabled the operator to change over the send-receive switch or to effect small adjustments in the tuning of the circuits.

A dual control equipment, comprising two microphones, two pairs of headphones and two coupling units, enabled either the pilot or navigator to use the wireless set or to converse with each other. Power for the anode circuits of the valves and for the filament lighting batteries is supplied by a wind-driven generator developing 100 milliamps at 1500 volts and 6 ampères at 7 volts. A trailing aerial approximately 200 feet long is used for the set.—"The Engineer," February 19, 1926.

METALCLAD RIGID AIRSHIP DEVELOPMENT.*

By Ralph H. Upson,
Chief Engineer of Aircraft Development Corporation, Detroit, Michigan.

A brief account of the accomplishments of present-day airships, with a statement of their known faults and their present limitations, indicates that until there is a radical change in design further advancement will be slight.

In 1897 an Austrian named Schwartz designed a sheet aluminum airship, the construction of which was actually finished after his death. It was inflated by fabric gas-cells and got into the air, but it could not be operated, due to serious faults in the structure, power plant, and controls.

Zeppelin's first metal-framed airship was also a failure, but his genius and perseverance won out in the end, and the duralumin-framed Zeppelin became the world's standard in airship construction. In a way it had too much success for the good of airship development generally, for although the Zeppelin itself was greatly improved by refinements of design and construction, it naturally discouraged further radical changes, especially during the War. Since the War, however, the very attractive possibilities of metal construction have proved irresistible.

Already the all-metal airplane is a practical reality, being made from the same duralumin that had previously been developed for the framing of airships. The success of the all-metal airplane is now in turn a great stimulus toward the elimination of dry goods as a covering for airships. The solution of this problem has involved a combination of:

(1) Inquiry into the general feasibility of the project.
(2) Search of materials and methods already available.
(3) Invention and development of means to fill the new requirements.
(4) Detailed calculation and experiment to establish a sound design.
(5) Similar research as to means of practical production.

Five years, representing about thirty man-years, have been spent on this program, which is now practically finished. The general feasibility of the Metalclad airship has been mathematically and experimentally proved. Sheet duralumin in quality now available proves immensely superior to fabric in almost every respect. Means have been devised for utilizing to full advantage the fundamentally superior qualities of the material. The stresses and aerodynamic characteristics have been most carefully studied for a great variety of operating conditions, showing a strength, stability, and general efficiency greater than any previous airship. Production methods and equipment in themselves have involved considerable new development to suit the requirements of the design. Successful research indicates that Metalclad airships, with all their other advantages, can ultimately be built more cheaply than the fabric-coveed type.

The first design to be laid out with any attempt at completeness was for a ship of 1,600,000 cubic feet. When this size was chosen it was with the idea that it was about the smallest Metalclad unit which could be made a practical success, the original thought being that some sacrifice in lift would have to be made to gain the practical advantages of metal construction. The first big surprise came with the weight statement of this ship, which showed an actual reduction in the total weight compared to fabric construction, with a strength and over-all efficiency substantially better than the somewhat larger *Shenandoah*.

The assumed duralumin plating weighed approximately four times more per square foot than the fabric cover of the *Shenandoah*. Yet the total

* Extract of paper delivered before Society of Automotive Engineers.

weight was less! The result was due largely to the manner in which the single metal surface was made to serve a number of different purposes, made possible by the homogeneous character of the structure.

Considered as a cover, the metal serves the same purposes as the fabric outer cover, but does it more efficiently by eliminating the flapping and moisture absorption so common to fabric. That is, however, only the beginning. In the Metalclad ship the surface plating also holds the gas and, in combination with the frame members, carries most of the stresses. The general principle of construction is similar to that of a steamship, in which the frame and plating are neither of them structurally self-sufficient, but each supports and reinforces the other. Thus the present design is far more than and essentially different from a mere metal-covered airship.

This saving in weight, made possible by the unit structure and by refinements in design, in turn brought up the question as to whether we could not produce a still smaller size ship for demonstration purposes. Study in this direction has finally crystallized into a size of only 200,000 cubic foot volume, about one-tenth that of the *Shenandoah*, and smaller than has been attempted before for any rigid airship. This demonstration unit, which is illustrated in the accompanying drawing, and is designated the *MC-2*, will have a better performance in most respects than that of a non-rigid airship or "blimp" of similar size. Thus although too small for economic commercial service, this ship will have characteristics of great value for scouting, training, and other special purposes. Tentative designs and performance calculations have also been made for ships of 80,000, 500,00, 1,200,000, and 5,000,000 cubic feet, respectively. But reports recently appearing in the press of the alleged *completion* of plans for the last-named size are, to say the least, greatly exaggerated. The 200,000 cubic foot size is the only one ready for immediate construction. In many respects this small size will present more conclusive demonstration of the Metalclad principle than could a larger unit in which the weight consideration would not be so serious.

A non-rigid hull tends to assume a shape distorted from the true circular shape, thereby reducing the fabric stresses. Such distortion would be fatal in a metal shell, but in this case, because of its rigidity, it can be built to the proper shape from the start. The shape is then independent of the internal pressure, and all stresses in the metal covering are simple tensile stresses with no tendency to buckle. However, it is advisable to maintain a light pressure inside to reduce vibration, and this is met by placing air scoops under the nose in the proper position so that at all speed the air flow at the scoops is proportioned to give the proper internal pressure. But if at any time, such as handling on the ground under high winds, a greater strength is needed, this can be accomplished by putting a relatively high pressure in the hull, which will stiffen it considerably.

A further investigation of the aero-dynamical properties of airships revealed that pressures due to motion through the air on the outside of the envelope were greater than atmospheric at the nose, less than atmospheric over most of the length along the side, and arose to positive values at the tail because of the "sucking in" effect there. For an object such as an airship hull it was found that the total resistance was only slightly larger than the skin resistance, indicating that the impact resistance was very low. This finding led to the design of a new-type hull.

Formerly it was thought that an airship had to be long and slim to go through the air easily and have proper stability. Directly or indirectly, this false assumption has been the guiding hand of airship construction for a quarter-century. Count Zeppelin based his design on it. The semi-rigid airship was created for it. The non-rigid airship was greatly complicated by it. Throughout all these years the airship has struggled with

the veritable Old Man of the Sea on its back. Even now, most engineers are only beginning to see that what success the airship has had has been in spite of this false god, who has claimed such heavy tribute in the wreckage of ships and lives. Wholly aside from their application to Metalclad construction, the improved aerodynamic and structural characteristics of the short hull, here described, are of fundamental importance.

For the Metalclad ship, good structural efficiency requires a fairly short and compact hull. We approached the problem confident that with the proper curves such a shape could be made efficient. The results are rather surprising. According to the best evidence available, the Navy wind tunnel, our new hull form has a lower resistance for equal volume than any shape hitherto produced. This is for a length diameter or fineness ratio of only 2.8 to 1.0 compared with 7.2 to 1.0 for the most recent *Los Angeles* and 8.6 to 1.0 for the *Shenandoah*.

The improvement in aerodynamic stability is even more striking. With our new fin arrangement totaling 17 per cent less area than the *Shenandoah* surfaces, for equal volumes, the stability and control are more than twice as good, and studies now under way give promise of still better results. This almost revolutionary improvement is largely due to the detail of the fin arrangement in which a greater number of small units, in this case eight, is used, instead of the four that became conventional about ten years ago. The static stability of the *MC-2* is somewhat excessive, but the larger Metalclad units with distributed loads and our internal corridors can be made about right. The aerodynamic lift is almost double that of the *Shenandoah* for equal volumes.

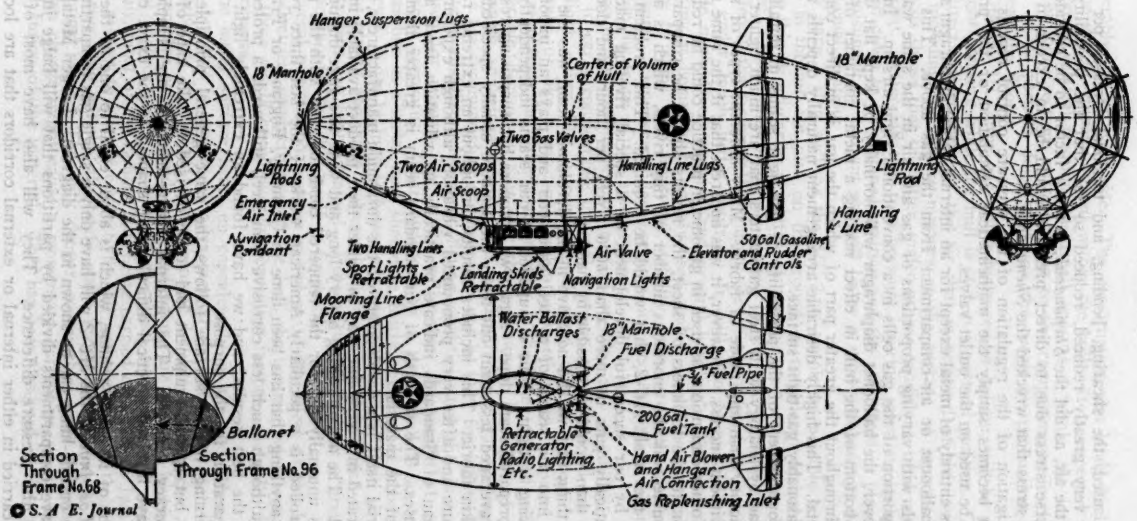
The next problem was to find some way of running an overload test on the hull itself. Even if desired, this could not be done on the full-size ship because its lift is strictly limited by its gas content and aerodynamic properties. Many tests had, of course, been made on individual girders, plating, and fittings, but a practical demonstration was also wanted of the structure as a whole. This was done by a hydrostatic or "water model."

To go into the general theory of an airship water-model is not necessary.* The reduced-scale model is simply hung upside down and filled with water whose weight acts in reverse proportion to the gas lift in the full-size ship, other forces being also applied in proper proportion. The usual scale of a water-model is the "natural" one of 1/30 the linear dimensions. This produces surface stresses just equal to those in the full-size ship. If the model is built to a larger scale than 1/30, using the same surface materials, it corresponds to overloading it, as if it were filled with a heavier fluid. For the metal water-model a size of 1/14 was chosen, which, when inflated to a corresponding pressure-head, is equivalent to overloading the full-size ship about five times and subjecting it to a pressure sixty-five times as great.

This is so severe that the effect of surface stiffness of the material is practically eliminated. In addition, the frame was reduced to the minimum and made relatively more flexible than in the full-size ship. In figures, the water-model hull, including its internal structure, weighs 51 pounds and holds over 2½ tons of water. It has not only the strength to carry this large overload, but is rigid under all conditions, including arbitrarily applied bending-moments that are far greater than any which could be expected in flight. In general the water-model results checked so closely with the calculated stress values that for future units it will probably be unnecessary to make water-model tests.

A technical discussion of the calculation of the reserve strength in the hull reveals that an actual safety factor of three is produced under the analysis of ten different conditions of loading. Because of the chunky

* See "Tech Engineering News," Vol. VI, No. 3.



DRAWINGS FOR 200,000 CUBIC FOOT METALCLAD AIRSHIP

The MC-2 is the smallest rigid airship ever attempted, and is to be constructed for demonstration and test purposes.

form to be used, the shearing, bending, and torsional stresses per unit volume are very greatly reduced, reduced so far that the possibility of breaking in the air, as did the *Shenandoah*, is not a factor to be considered. The transverse stress due to direct pressure would be in excess of the longitudinal stress that wrecked the *Shenandoah*.

An investigation of the regulation of the pressure control reveals that the problem becomes simply the mathematical calculation of the proper diameter to be used on the outlet valves.

The hull is entirely of metal except for an internal fabric diaphragm separating the balloonet or air-compartment from the gas above. This diaphragm yields with varying proportions of gas and air in the same way as the bottom portion of the gas cells in a conventional rigid airship. In the *MC-2*, however, the balloonet diaphragm will normally be kept flat down against the bottom of the hull in effect making a metal container of the entire hull throughout the greater part of which the gas is in direct contact with the metal. This highly desirable arrangement naturally depends upon having a reasonably gas-tight surface.

Early encouragement of the possibility of gas-tight seams in duralumin sheet was had by analogy with steel gasometers that are much tighter and more satisfactory than any made of fabric. Many different types of seam were tried without success. However, it was apparent that if the same rivet spacing and other dimensions as used in gasometer practice could be reduced in proportion to the thickness of sheet, the results should be comparable. The big trouble was the enormous number of tiny rivets, which is about 3,000,000 in the small *MC-2*, that would be required. This problem has been solved by our successful development of a special riveting machine that automatically puts in more than 5000 rivets per hour, and does it much better than would be possible by hand.

The only thing that we did not have in the duralumin seam was the rust that works into the seams of a steel gasometer and plays an important part in making it tight. This is supplied to our seams independently by a specially prepared seam dope. Again we attain rather surprising results. Tests have averaged less than one-tenth the leakage usually specified for goldbeater-skin fabric. This includes results throughout an extreme range of temperature and after very pronounced vibration, although experiments at the Bureau of Standards indicate practically no vibration from aerodynamic causes. The strength of our standard seam is greater than the yield-point of the material.

The material itself is a development of the duralumin manufacturers, who have co-operated in a very fine way to render their product available in the form needed. The art of rolling a very long and wide fine-gage duralumin sheet that is unusually flat for the tempered condition, and with the gage closely controlled, is a peculiarly American work of the last three years, and beyond anything that has been done in Germany, England, or France. Duralumin, although much less corrosive than steel, still needs a protective coating for the best results; and we have found an extremely light and efficient preparation for the purpose.

The patterning of the surface follows the same general principles as with fabric, but special equipment had to be devised to take care of the greater accuracy required. The internal frame-members are simple in form, but have involved practical difficulties of shaping to the exact curve and angle.

On account of its small volume, which is about equal to one of the gas cells in the *Los Angeles*, the *MC-2* will have only a single gas compartment, and a single outside-hung car as shown in the diagram. Larger Metalclad units will have compartments divided by partitions that will bulge in the direction of the pressure difference. They will also have most of the useful load carried in either internal or external corridors that are located

preferably one on each side. The construction also lends itself admirably to the use of engines housed within the hull. It is expected to erect the hull in a vertical position, the small ones as a whole, the larger ones in two or more sections corresponding to the compartments in the completed ship. This permits doing most of the work by production methods on or near the floor.

With the *MC-2* the gas, either hydrogen or helium, will be put in under a slight pressure while the hull is still in a vertical position. This is done by direct displacement of the air that is removed at the lower or stern and including the impure mixture at the surface of separation. Careful analysis shows that the latter, under proper conditions, need not exceed ten per cent. After inflation to a pressure insuring extra strength, the hull is hauled down to its designed position and the car bolted in place. Ordinary replenishment of buoyant gas will be by a service connection with the hull in normal position.

To have a ship such as the *MC-2* as simple as possible to handle, not only in the air but also on the ground, is considered desirable. In both cases this is primarily a matter of aerodynamic stability, as before mentioned. These improvements applied to the *MC-2* mean that it can be handled in the air by half the usual crew for a ship of its size, at the same time under much better control. It will also be possible to use very simple methods of mooring and ground handling. The principle of bow mooring is bound to be widely used, and it should be particularly appropriate to ships with improved stability, extra strength, and resistance to exposure. Our recent developments in mooring-tower construction are outside the scope of the present paper.

Due to its combination of perfect rigidity and its ability at the same time to carry considerable internal pressure, the Metalclad can be operated in flight either like a conventional rigid airship without reference to pressure, or like a non-rigid airship by watching the pressure manometer, and regulating the altitude accordingly. The latter method will be usually preferable, because in that way the hull can be kept full of gas and the maximum advantage taken of its extremely good gas-holding properties.

We are recommending the hydrogen gas be used for inflation on account of its availability, cheapness, lifting qualities, and the fact that it can be used for reserve fuel, which will reduce the weight of water ballast recovery apparatus. That hydrogen in a metal hull is at least as safe as gasoline in a metal tank should be fairly obvious. If helium is desired, however, it can be used to better advantage than in a fabric airship because of the almost negligible leakage through the metal hull, and the higher gas purity that can be maintained. Even with helium it is a great asset that the surface of the ship itself is fireproof.

Several questions have been raised as to the effect of the high thermal and electrical conductivities of the hull, the common idea being apparently that these would be unfavorable operating-factors. Actually they are just the reverse.

The metal surface heats up considerably in the sun, when not in motion, but this is usually an advantage at the start. In operation, thermal disturbances in the lift are eliminated by keeping the gas at the same temperature as the outside air. Indications are that the highly conductant metal surface tends to approach this ideal when in rapid motion through the air, most of the radiant heat being carried off again as fast as received.

In respect to electric conductivity, all authorities seem to be agreed that the new ship will be absolutely static proof and almost, if not entirely, lightning proof. The only danger conceivable from lightning is that a sudden charge might cause a momentary current so severe as to produce local fusion of the material. This has been observed in the case of wires

where the current is restricted to a one dimensional flow, but for a flat surface it seems at least reasonable to suppose that any charge would splash out over enough area to prevent melting through.—" Tech Engineering News," April, 1926.

THE GOODYEAR SEMI-RIGID AIRSHIP *RS-1*.

The semi-rigid airship *RS-1*, built by The Goodyear Tire and Rubber Co., of Akron, Ohio, for the U. S. Army Air Service, has successfully passed through its recent test flight period, and is now listed for active flying duty at Scott Field army post near St. Louis.

The first American-built semi-rigid airship, it is 282 feet in length, 70 feet 6 inches diameter, and has a gas volume of 720,000 cubic feet. Powered with four low-compression Liberty engines in two streamlined power cars located aft and suspended from the keel, the *RS-1* develops a total of 1200 horsepower, giving a maximum speed of over 75 miles per hour, under favorable conditions, and a cruising speed of 45 miles per hour, making it the fastest as well as the largest airship in the U. S. Army Air Service.

Construction of the parts for the *RS-1* commenced in the Goodyear factory at Akron towards the end of 1924, and the parts were then taken to Scott Field for assembly last spring. Although the ship was completed late in 1925, a slight mishap caused by a mistake made by a rigger during erection, prevented trials being carried out until early this year. On its initial trial flight, on January 8, it made a successful trip of over an hour's duration in a mild snowstorm, carrying a crew of eight men and with Lieut. Orval Anderson in charge.

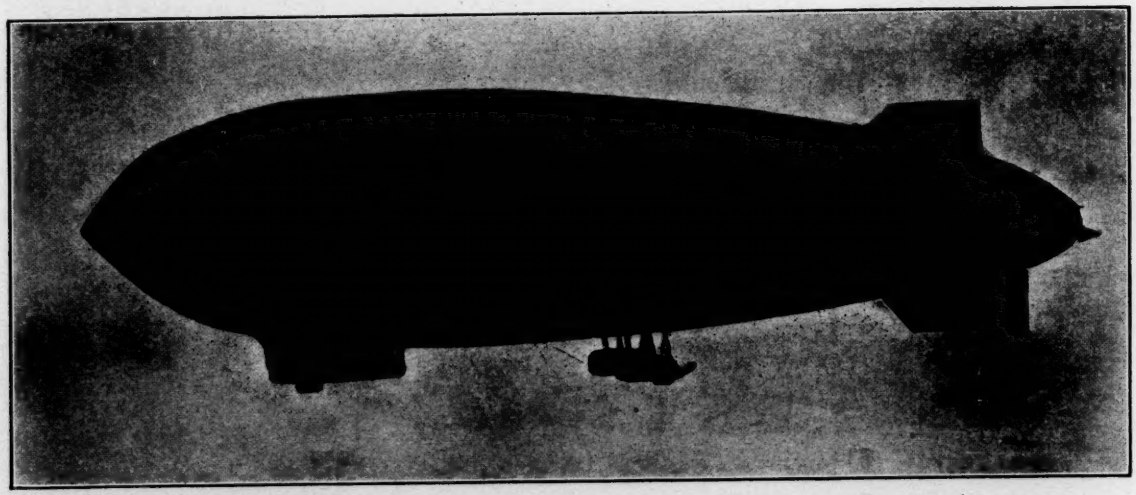
During a subsequent test flight the *RS-1* encountered extremely bumpy flying conditions and winds that at times exceeded 50 miles per hour in velocity. After a flight of 19 hours' duration, however, the *RS-1* was landed at its home base in a high ground wind without mishap, the personnel rendering a favorable report of the ship's entire performance.

The *RS-1* resembles in many ways the Italian semi-rigid airship *N.1*, previously described in "Flight," although the American ship is slightly larger and is inflated with helium.

Along the belly of the envelope is an enclosed aluminium alloy keel, suspended by catenary cables, which attach to the three points of the keel at 10-foot intervals. The load coming on the center catenary gives the envelope a heart-shape transverse cross-section, of which the greatest dimensions are 66.5 feet by 73 feet high. The envelope is divided into four compartments by transverse diaphragms. The center diaphragm is a solid wall, while the end ones have surge holes near the bottom in the gas and air compartments. The air balloonets, with a total capacity of 224,000 cubic feet, are also divided into four by these diaphragms.

The nose of the envelope is provided with 19 duralumin battens secured to the keel extremity, laced to patches on the envelope and supported laterally by five sets of brace wires and two telescopic rings of duralumin tubing. The nose is also provided with mast mooring equipment.

The keel, which forms a great vertebrate spinal column, is constructed of duralumin columns, Phœnix type, of a maximum length of 10 feet. These columns are connected by a ball and socket joint in a forged lynite housing, which housings are part of the rigid transverse triangles. The tension wires of the keel are of streamline form. The fuel, oil, and water ballast tanks, and fore and aft drag ropes are suspended in the keel from the transverse keel frames.



THE GOODYEAR "RS-1" AIRSHIP; THIS IS THE FIRST SEMI-RIGID AIRSHIP TO BE BUILT IN AMERICA.

A control car, 35 feet long and entirely enclosed, is suspended from the keel under the nose of the ship. This car contains the navigating controls, sleeping facilities for officers and men, a motion picture camera and a radio installation.

It is constructed of duralumin framework with fabric and metal sheathing. The control room is separated from the rest of the car by a hollow partition which houses the control cables coming down from the keel. The equipment of the pilot car, in addition to the latest navigating instruments, includes an SCR—135 radio set, a type B-3 camera, bomber's cockpit and equipment, four berths, an auxiliary power-driven 5000-cubic foot capacity air blower, map table, etc., etc. The car weighs, 1900 pounds, exclusive of radio, camera, and bomb sight.

The armament includes bomb racks of 3500-pound capacity and single machine-gun mounts on both sides of the forward end of the pilot car; a gun mount can easily be arranged for at the extreme bow of the ship, if required.

Owing to the fact that the *RS-1* is the first of the semi-rigid type to be built in America, it has been designed to have a high factor of safety, under all conditions, performance being a secondary consideration if it interfered with adequate strength.

It has been designed for training, photographic, and patrol duties, and will later be equipped with a newly designed suspension device to pick up and release a small aeroplane during flight—a development with which the U. S. Army Air Service has experimented for the past three years.

Although considerably smaller than the American rigid airships—the ill-fated *Shenandoah* and the *Los Angeles*—the *RS-1* has been designed so that it can be moored to the masts erected for the rigids. It may be mentioned in conclucion that the *RS-1*, while the first airship of the semi-rigid type constructed in America, belongs to a long line of lighter-than-air craft turned out by the Goodyear Tire and Rubber Co., which some little while back acquired the patent rights of the German Zeppelin Co. for North America.—"Flight," March 25, 1926.

NON-CORRODIBLE NICKEL-CHROMIUM STEELS.

Among the metallurgical developments of the last year, mentioned in the January number of "The Metallurgist," an outstanding one was the increasing use made of the austenitic non-corroding steels. It has long been well known that the resistance to corrosion of ordinary rustless steels could be greatly improved by increasing the chromium content, but the use of higher chromium was not practicable, since it led to the production of strongly air-hardening steels. Chromium steels of the ordinary rustless type, but with more than 15 or 16 per cent of chromium, are permanently hard and non-machinable. They are not susceptible to ordinary heat treatment and cannot be softened, since they retain their martensitic structure, even after very slow cooling.

An analogous effect is produced by increasing the nickel in a nickel steel. As the nickel content is raised the steel shows increased resistance to corrosion, but with more than 10 per cent of nickel a low-carbon nickel steel becomes permanently martensitic and commercially of little value. When, however, the nickel content reaches 22 per cent a steel is produced which consists of austenite, *i. e.*, the solid solution of carbon and alloying elements in γ iron, and for this there is a new field of utility. The properties of the austenitic nickel steels have long been known, but recent developments spring from the observation that a martensitic chromium steel can be converted into an austenitic steel by the addition of a small per-

centage of nickel, and a martensitic nickel steel can be made austenitic by the addition of Chromium. To produce a low-carbon austenitic steel containing 20 per cent of chromium, a nickel content of at least 6 per cent is required. Similarly, a 15 per cent nickel steel requires about 5 per cent of chromium, though, unlike chromium, nickel alone in sufficient quantity will render a steel austenitic.

We thus have two groups of steels, the high-nickel or high-nickel low-chromium steels and the high-chromium low-nickel steels. A good deal of information about them is to be found in the publications of Sir Robert Hadfield and Dr. W. H. Hatfield, and in the contributions to the symposium on "Heat and Corrosion Resistant Steels," held by the American Society for Testing Materials in 1924. From the published data it is clear that, though both classes of steel are resistant to alkalis and to many salts, the high-nickel steels are particularly resistant to the action of sulphuric and hydrochloric acids, and to salts, such as magnesium chloride, which are partially hydrolysed in solution, while the addition of nickel to the 18 to 20 per cent chromium steel does not diminish, but may rather increase, its exceptionally high resistance to corrosion by nitric acid, sea water and the atmosphere. The high-nickel steels are not resistant to oxidizing agents or to atmospheric oxidation in the presence of sea water and other salt solutions, unless they contain a considerable amount of chromium. Thus we get a third group, which consists of high-nickel high-chromium steels, though the chromium content is not as a rule as high as 18 per cent. These steels are more expensive than those of the other two groups, but they probably have a more general all-round resistance to the action of different media.

Another characteristic of high-chromium steels which is retained when nickel is added is their resistance to scaling at high temperatures, but they are stated to lose strength rapidly when the temperature exceeds 700 degrees C. In the nickel-chromium series of steels the composition which probably retains its strength best at high temperatures is one with a total nickel plus chromium content of about 50 per cent, the greater part of this being nickel. This indicates another application of steels of the third group defined above. It seems likely, however, that the plain nickel-chromium steels may be less satisfactory in this respect than more complex steels containing tungsten, or other special alloys of iron, chromium and nickel containing 20 per cent of chromium with 50 per cent or more of nickel.

A few words may be added with reference to the general properties of these austenitic steels. They are soft, and may be stamped, pressed, rolled or drawn cold, but they harden rapidly under cold work. They are easily forged and rolled hot, and may be welded by any of the usual methods. In the soft condition they have a very high elongation and impact figure, but the limit of proportionality is low and the yield point is barely 50 per cent of the ultimate tensile strength. These mechanical properties cannot be altered by the usual methods of heat treatment. Working at temperatures just below redness, or cold working followed by annealing within a similar range of temperatures, improves the yield point of these steels to a marked degree, without reducing their resistance to corrosion. Most of these steels are non-magnetic, they offer a high resistance to abrasion, and they are all difficult to machine, some of them very difficult. They may be burnished, and retain a high degree of reflectivity under severe conditions of exposure; but, for their resistance to corrosion, they are not so dependent on surface finish as are the ordinary rustless steels.

In selecting one of the many non-corroding austenitic steels now on the market, the particular purpose in view—for example, the medium to which the steel must be resistant—is naturally the first consideration, and makers' pamphlets will generally be found to give full details of the character-

istic properties of each steel. Unfortunately, the advent of these steels has led to a widespread revival of "trade names" to define the products of different firms. When the mystery of these fanciful designations has been penetrated, the choice is usually limited to one of the groups mentioned above, but sometimes steels from two of the groups may be equally applicable. Then, as a rule, the deciding factor, in conjunction with cost, will have reference to mechanical qualities—not tensile properties, which are approximately the same for all these steels, but such properties as ease of machining or of cold working, cupping, stamping or other preliminary treatment which may be necessary before use can be made of the steel.—"The Metallurgist—Supplement to the Engineer," 26 February, 1926.

THE NAVY AND THE STEEL INDUSTRY.

By G. K. Spencer.*

Great changes have been brought about in naval warfare by the introduction of various aeronautical and chemical weapons. This renders it essential that all battleships be accompanied by light cruisers and submarines for primary operations, and by destroyer leaders for handling destroyers in the face of the new weapons. New construction desired is described by Admiral Charles Hughes, commander-in-chief of the battle fleet, as "vitally necessary, if we are to move successfully from our own shores in the event of hostilities."

Laymen in naval affairs are generally almost totally unaware of the widespread changes in tactics involved by the new weapons. A large ship can be blinded, unless adequately protected by an extensive force of cruisers, submarines and aircraft, through aerial and chemical operations. In recent tests a ship was bathed in chemical fog, while air forces "bombed" it to death, its gunners unable to see the foe.

The proposal is now openly broached that all United States merchantmen be equipped, in event of hostilities, with aircraft, catapults and aircraft personnel. These would be provided much after the system which we utilized in providing merchantmen with naval gun crews during the late war. This, again, involves the construction of a great reserve supply of catapults, which may speedily be mounted on merchantmen.

Immediate need is stressed for 22 new 10,000-ton light cruisers, 18 destroyer leaders, 23 large fleet high-seas submarines, at least one new aircraft carrier, four new squadrons of aircraft, an extensive number of catapults for delivering air craft into the air, faster supply vessels—and 10,000 more personnel to handle the new units.

In solving these new and modern problems the element of armor plate is not a major, though an important, factor. Rather, the development of the gear already alluded to as necessary for the proper balancing of the fleet, and the evolving of armor-plated amphibious tanks for both water and land fighting, will assume the major importance. The Navy already possesses a small squadron of amphibious tanks, for purposes of landing, on enemy shores, forces from the fleets at sea. The day when lines of

* San Diego, Cal.

From the earliest steel ships, begun in 1883, to the present writing, about 2,185,000 tons of steel and iron—mostly steel—has gone into the construction of United States naval vessels. While this is, perhaps, one-fourth of 1 per cent of the enormous tonnage of rolled steel produced during the 43 years, yet its influence on the development of the steel industry has been out of all proportion to its tonnage.

The article here presented traces briefly some of the development features in which the Navy and American industry have gone hand in hand. It has been claimed that without this impetus of naval needs the American steel industry would not today be in the commanding position which it enjoys. This article shows why.

men might be landed from open boats has passed, as demonstrated in the debacle at Gallipoli, and in recent maneuvers of our own fleet. Industry will be requested to cooperate in the further evolution of heavy machinery of all types for the new construction, as well as for ordnance.

Studied movements of sea warfare of the past have given way to a furiously speedy movement. Smoke and gas and fog are liberated by aircraft and surface vessels, to shield any number of ships or aircraft or individual units. Colored fogs which simulate the color of the horizon obliterate all trace of the forces they protect. Highly specialized vessels are required, quickly and expeditiously to search out these feints and shelters. Forces unprepared to cope with this kind of fighting, even if they are extremely powerful potentially, can be destroyed piecemeal by much inferior forces, adequately prepared for these tactics.

It is this subtle point which the naval chiefs are trying to indicate to the country and to Congress. Naturally, Congress lags in an understanding of these new aspects of war at sea, for the whole picture is so radically different from what it was even three years ago. To a considerable extent, the development of the light cruiser, the destroyer squadron leader and the aircraft—as well as the powerfully gunned scouting submarine, of which we have but three, is responsible for this change. And the United States has been left far behind, not because of naval reticence in bringing the facts before Congress, but because of the natural difficulty in bringing laymen to understand the problem.

Congressmen recently have been accompanying the fleet maneuvers and instruction has been given them in these new factors. The *Henderson,* a large naval transport, is used for this instruction, carrying to each naval maneuver a party of interested Congressmen, and members of the naval affairs committees of the House and of the Senate.

THE NAVY'S INFLUENCE ON THE STEEL INDUSTRY.

The influence of the Navy on the development of the steel industry in the United States has been considerable. Steel plates were first used in this country for making an experimental firebox for a locomotive, at the Altoona shops of the Pennsylvania Railroad, in July, 1861. Not until 1865 was a second firebox made of this material. The difficulties in making plates from steel were great, as steel could be produced only by the crucible process. Problems in connection with making steel plates were, however, sufficiently solved, so that by 1880 such plates were being used more or less generally for locomotive boilers and fireboxes. As the product was still so uncertain that manufacturers would not accept orders for material conforming to specifications, the purchaser had to depend for quality on the general guarantee of the manufacturer.

In 1869 steel was used in the construction of the St. Louis bridge across the Mississippi River. Its use was considered such a bold piece of engineering that the experiment was not repeated until about 1879. Then Bessemer plates and shapes were adopted for the roadway, girders and approaches of the East River (Brooklyn) Suspension Bridge. The use of steel for shipbuilding had made practically no advance by that date, although one establishment, the Pusey & Jones Co., Wilmington, Del., had constructed five small river steamers and six lighters with steel plating, most of which was imported.

In 1881 Secretary of the Navy William H. Hunt appointed a board for recommending the design and construction of new vessels for the Navy. The work of this board marks the beginning of both the modern Navy and of the manufacture of steel as an industry in the United States. The board made an exhaustive report on the types of ships that should be built and also recommended that they be built of steel. In the latter respect the

recommendation of the board was not unanimous. A minority report, by the more conservative element, was against the use of steel, as it was still considered a highly experimental material.

At that time, in the opinion of these members, steel of satisfactory quality could not be produced in the United States. Finally, however, the views of the majority prevailed, and four vessels built of steel of domestic manufacture were authorized by Congress. These vessels were the *Atlanta, Boston, Chicago* and *Dolphin*—the A, B, C, D of the "new Navy." The contract was placed with John Roach, Chester, Pa., on July 26, 1883.

At this point the real influence of this building program on the steel industry of the country begins, not because of the tonnage involved (about 10,000 tons of plates, shapes, forgings and castings), but because of the prescribed specifications as to quality. At first no steel manufacturer was willing to accept a contract to deliver steel in accordance with the Navy Department specifications. Finally, John Roach himself built a mill, to make certain of the material. The mills which supplied the steel were the Chester Rolling Mills, Chester, Pa.; Norway Iron & Steel Co., South Boston, Mass.; Park Brothers & Co., Philadelphia, for angles, tees and deck beams.

The difficulties which these mills experienced in producing steel complying with the specifications were enormous. Great pressure was brought to bear on the Navy Department to modify the requirements, but all such efforts were successfully resisted. Finally, material was produced which met the specifications and, what was of greater importance to the steel industry, manufacturers were taught how to make good steel. From that time on there was no question that the United States could produce steel as good as any abroad.

The *Dolphin* was completed in 1885, the first of the building program to be delivered. This vessel, after 36 years of service, was placed out of commission at the Boston Navy Yard in December, 1921, and was sold in 1922. No better testimonial to the quality of the material which entered into the construction of this vessel is needed than this long record of service.

One of the most difficult and expensive problems encountered by John Roach, when he undertook the building of the first ships of the new steel Navy in 1883, was the construction of the stern frame for the *Chicago*. At that time this important part of a ship had to be forged, as the steel-castings industry, scarcely even in its infancy, could not supply castings for this purpose. Only small parts were made of cast steel and the engineering profession had no confidence in this material.

One day in 1885 the Secretary of the Navy, William C. Whitney, visited Roach's shipyard in Chester, Pa., where the *Chicago* was being built. In discussing the great difficulty of making a satisfactory stern frame forging for the vessel, Naval Constructor Lewis Nixon (later, the designer of the *Oregon*) suggested to Mr. Whitney that stern frames on future vessels be made of cast steel. This was a bold suggestion, as no structural steel castings of such size had been attempted before.

As a result of this conversation Mr. Nixon was detailed to investigate the possibility in the future of making such parts of cast steel. No steel foundry at that time had either melting capacity or annealing ovens of sufficient size to handle such large castings. However, the foundrymen had confidence in their product, and were anxious to make the attempt, offering to increase their equipment as necessary to produce stern frames.

After much opposition from the more conservative engineering element, both in and out of the Navy, the stern frame for the *Charleston* was specified by the Bureau of Construction and Repair to be made of cast steel. Much difficulty was experienced in making and annealing the casting. Finally, in 1887, the Midvale Steel Works delivered a satisfactory casting

for the purpose. This piece weighed 6720 pounds and may be said to mark the beginning of the cast-steel industry, because of the influence it had in bringing the railroads around to the use of such material.

LOCOMOTIVE FRAMES OF CAST STEEL.

This phase of the development of the steel-castings industry deserves some additional comment. At the Chicago Exposition, in 1893, Krupp exhibited some steel castings which at that time were considered remarkable, among them being a cast-steel frame for a Pennsylvania Railroad locomotive. At a meeting of the American Society of Mechanical Engineers doubt was expressed as to whether any American steel foundry could make such a casting. It was then pointed out that the Navy was using steel castings for stem and stern frames of naval vessels; that these castings were, with each class of ship, increasing in size and complexity of shape; and that they were satisfactorily passing the Navy Department tests, which were well known to be severe and guarantees of good material.

With the Navy leading the way, the railroads gradually swung around to the use of steel castings for locomotive frames and other structural parts. There is no doubt that the remarkable development of railroad locomotives and rolling stock in the United States can be traced in great measure to the adoption of steel castings for locomotive frames and other parts. The difficulty, cost, and perhaps impossibility of making locomotive frames by the forging process, of such sizes as are now used, would have set a limit on this development.

It is interesting to note that in 1883 the total production of steel castings in the United States was only 1684 tons, and that this output has risen to almost 2,000,000 tons annually. The Navy by its early example stimulated and encouraged the steel-castings industry and in turn advanced by many years the use of such castings by the railroads.

DEVELOPMENT OF THE STEEL INDUSTRY.

The history of armor development alone is of the greatest interest. Prior to 1890 there were no plants in this country capable of turning out heavy forgings of either wrought iron or steel. Our heavy forgings were produced abroad. Many attempts had been made to forge large shafting under the hammer—the only means available at that time—but the product was not satisfactory, for the thick masses of steel could not be penetrated by the blow of the hammer. Cracks could not be eliminated in the interior and the forgings were naturally unfit for use.

With the improvement in range and hitting power of heavy guns came the necessity for defense, and armor plate was the logical resort. The world was searched for methods of manufacture of suitable armor plate. In this search and subsequent laboratory investigations of the effect of alloying steel with various other ingredients, and of novel heat treatments of steel, there were discovered or evolved many of the special alloys of iron with nickel, chrome, vanadium and other metals which today are of the utmost importance to industry in general. The Bureau of Ordnance paid for many of these experiments, allowing great laboratories to be supported, thus permitting this most important work to be carried on to the highest degree of success.

As a result of continued experiment in the production of high-grade steels suitable for Navy purposes, an impetus was given to this industry which has carried it along until it stands at the head of the steel industry of the world. Not only does the American steel industry produce steel of the highest grade for every purpose, but it may be said to be the final arbiter in the matter of production and cost.

In the latter half of the nineteenth century Sir Joseph Whitworth of England invented a hydraulic forging press. Unsuccessful attempts were made to obtain the use of his patents in this country. An accurate description of the press could not be obtained and strangers were not permitted to visit the Whitworth shops, nor would Sir Joseph make a press for use in this country. In 1883 the Navy Department sent to Europe a gun foundry board, of which Lieut. William Jaques was secretary, for the purpose of visiting iron and steel plants. Arriving in England, the board, after considerable difficulty, was permitted to visit the Whitworth shops, and Lieutenant Jaques obtained from Sir Joseph a contract giving Jaques, personally, authority to build a plant in the United States, the Whitworth Works furnishing the plans.

FIRST ARMOR PLANT IN AMERICA.

After the board returned home, Lieutenant Jaques took up with John Fritz, general superintendent of the Bethlehem Iron Co., the question of building at Bethlehem the forging plant for which he had contracted with Sir Joseph Whitworth. Mr. Fritz, much interested, after considerable difficulty persuaded the directors of the Bethlehem Iron Co. to undertake the project. The result was the first armor plant in this country.

While construction was under way, Lieutenant Jaques and Mr. Fritz made frequent trips to Europe to visit the Whitworth and other plants. In France in 1887 they obtained from the Creusot Works a contract permitting them to manufacture certain armor in accordance with the most advanced methods of armor manufacture then in vogue. On June 1, 1887, the United States Government placed its first contract for armor with an American concern, the Bethlehem Iron Co. This company, then a small operation, expanded greatly, finally becoming the Bethlehem Steel Corporation. The variety of its products for commercial use is too well known to require enumeration.

A short time after the first armor contract was placed Commander Folger, then Chief of the Bureau of Ordnance, visited Bethlehem and pursuaded the company to undertake the manufacture of gun forgings much longer than any that had been made previously. For several years forgings for tubes for 6-inch, 8-inch and 10-inch guns had been purchased in England. The first large caliber guns made entirely of American steel were those for the battleships *Indiana*, *Massachusetts* and *Oregon*, manufacture being started in the early nineties.

The quality of forged steel produced by armor plants has steadily risen under pressure of more exacting ordnance specifications from an elastic limit of about 35,000 pounds in 1887 to the homogeneous nickel-steel gun forgings of today, which are treated to obtain 60,000 pounds elastic limit, even in the largest forgings.

For many years forged steel used for certain ordnance purposes was required to have great strength and maximum ductility. As the forging grew in size, cost of production mounted. In the effort to combine safety and efficiency with economy, the Bureau of Ordnance experimented in the use of steel castings to supplant many of the forgings in use. As the result a high grade of cast steel was developed and castings of great weight were made.

SPECIAL MACHINERY AND RESEARCH REQUIRED.

To satisfy ordnance requirements in the manufacture of enormous gun forgings, thick, heavy armor plate, and large steel castings special machinery had to be designed. Naturally, the cost of development was borne by the Bureau of Ordnance, but in the end the nation as a whole benefited. Hydraulic forging machinery with high-power hydraulic pumps was devel-

oped. Machine shop tools of larger size than had ever before been attempted in this country were made. Powerful traveling cranes and conveying machinery were made and first used in connection with the manufacture of ordnance material. Means of casting and handling large unit masses of steel in ingots were developed. Large plate rolling mills for the armored decks of battleships were placed in operation.

Chemical research was necessary in the development of steel for armor, projectiles and guns. This has not only benefited the Navy in the improvement of its ordnance material, but has given to the nation steels which otherwise would have been necessarily retarded in development.

With a view to producing metal-cutting tools which would economically machine heavy armor plate and gun forgings, the Bethlehem Steel Co. developed the well-known "Taylor-White" process of treating high-speed steel. This steel is now used in metal-cutting tools that, by many, are considered as having revolutionized machine-shop practice throughout the entire world.

In research work, in the chemical and physical characteristics and properties of steels, and in the installation of heavy equipment necessary to manufacture heavy ordnance material, it may be safely stated that no steel plant would have ventured the extensive pioneer work without the prospect of sufficient Government work to justify it. In past years there was not enough demand for large commercial products of size sufficient to pay a profit on the investment, regardless of whether or not some of these products are somewhat in demand. However, since the requirements of the Navy have resulted in the design and manufacture of heavy equipment, the commercial world has been able to profit.

HEAVIER ROLLING STOCK DEVELOPED.

The demand of the Navy for suitable ordnance material has made its presence felt in the field of transportation. The ever-increasing weights severely taxed the capacity of railroads, and cars have been designed, outgrown and redesigned, primarily for the purpose of transporting guns of large caliber. In this field the Bureau of Ordnance has been the pioneer. At the time when cars of 80,000 pounds capacity were in common use, special cars had to be constructed of greater capacity to transport naval guns from the steel works to the navy yards. Cars of 200,000 pounds capacity have been built for naval purposes, and the average capacity of ordinary freight cars has gradually increased, to the cars of 120,000 pounds capacity of the present day.

Acid open-hearth steel is generally considered more suitable for gun and other high-grade forging than basic steel. The American ore used for the manufacture of pig iron necessary for making acid steel has not been plentiful in this country. As a result of the shortage of suitable American ore, the Bethlehem Steel Co. opened mines in Cuba and South America and built steamers to carry ore from the mines. That company now has in operation steamship lines which not only carry ore, but are used for other commercial purposes, a valuable addition to the American merchant marine.

To wage a sea battle successfully, it is necessary that the commander-in-chief bring his ships to the place selected. Once there, he must be able so to maneuver them that his guns can inflict the greatest amount of damage to the enemy, with the least possible damage to himself. It thus requires that each ship be absolutely reliable in respect to her ability to carry out her part of the program. Any other condition might cause such a disarrangement of plans as materially to change the outcome of the battle, with possibly disastrous results to the country's future.

It can thus be seen how extremely important it is for the machinery, which enables the ship to maintain her proper place in the battle or scouting line, to be as perfect as it is humanly possible to make it. We may well remember the old story of the battle that was lost for want of a horseshoe nail.

Various bureaus of the Navy contribute their part to this handling of the ships. The Bureau of Ordnance in general takes care of the operation of the guns; the Bureau of Construction and Repair of the ships' control, steering, etc.; the Bureau of Engineering supplies the power that propels the ship, as well as the various auxiliaries that furnish power for handling the guns and steering the ship.

The Bureau of Engineering is technically responsible for practically all of the power supply furnished on board ship. In the popular view, this applies more specifically to the power for moving the ship, and this is its most important single item. But along with this are numerous auxiliaries which contribute, each in its own way, to the efficiency of the whole ship.

As in any organization, business or sport, cooperation is the essential method without which no collection of men or machinery can perform efficiently. On board ship, confronted with the constricting requirements of space, such cooperation is particularly necessary. The development in naval engineering which this country has seen in the past three decades has been possible only through the utmost cooperation between those who design and supervise the construction of the machinery of the ships and the commercial firms which build and install the machinery.

STEAM NOT A WELCOME INNOVATION.

The progress of steam machinery on naval vessels had a hard birth and a bitter fight for existence in its early youth. Captains intimately familiar with their sailing ships loved to maneuver them, and were loath to transfer that personal contact to the engine rooms. As with any new development, the early motive power was not very reliable and breakdowns were frequent. All this increased the distaste for the new type of propulsion. Moreover, in the period of retrogression which followed the Civil War, money for development work at home was hard to obtain, and the initiative passed to foreign ship and engine builders.

With the passage of the act of August 5, 1882, which provided for the first vessels of our "new" Navy, began that enormous growth in engineering which culminated in the projected plans of six battle cruisers. Built into the hulls of 33-knot ships was to be machinery of upward of 180,000 horsepower—more than is found in any but the largest power plans on shore.

But the feature of this act of 1882 which was of the greatest importance, and had most far-reaching results, was its provision that the steel to be used should be of "domestic manufacture." It is hard to realize that, when this bill was passed, the shipbuilders of this country used either wrought iron or steel imported from England. Small as were the castings or forgings in those days, this country did not posses the necessary means for making them. And the designers of these early years recognized the necessity of incorporating nothing but the best of materials possible to use.

From this beginning the Navy Department, through all of its bureaus, has been in the forefront of industrial research and development. The cooperation with manufacturers, soon established, has been constantly maintained, the common aim being always toward the production of some more suitable material, whether for reasons of strength, life or economy. Now the problems of the Navy are in no wise different from those of any other great manufacturing concern, except that no tangible return upon the investment need be made. The Navy's value is potential, as in fire insurance, and until it fails the consequences can not be predicted.

EVOLUTION OF NAVAL MACHINERY.

When Congress authorizes the construction of a ship, it specifies a certain tonnage, speed and, possibly, armament. Of the tonnage, after numerous conferences, a definite weight and space are allotted the Bureau of Engineering wherein to place the machinery which shall fulfill the requirements demanded of the ship. This is the province of the designers. The evolution of naval machinery has been remarkable.

Commencing with cumbersome machinery, the best that could be produced, but heavy and complicated, the world has seen the evolution of the compound engine, then the multiple-expansion engine, then the turbine, to which later was added mechanical reduction gearing, finally culminating in the adoption of turbo-generators and electric motors for the main motive power. Each step was taken with the idea of increasing efficiency and reliability. And it has been only through the utmost cooperation of the firms manufacturing this material that such results have been accomplished.

At first suitable steels were not available. The bureau adhered rigidly to its specifications, which had been drawn so as to obtain the maximum strength of each part. The manufacturers could not meet these specifications but, with the aid and cooperation of the bureau, better steel and cheaper steel has been produced than was dreamed of years ago.

In following out the bureau's practice the manufacturers have increased their own earnings, as people will always pay for a better product, knowing that the initial cost is not the only cost that enters into calculations. To forge and machine this material in the large sizes necessary, larger, stronger and more accurate machine tools were required. These were gradually provided, until now the manufacturers of this country have no hesitancy in competing anywhere in the world.

INSPECTION OF MATERIAL ASSUMES HUGE PROPORTIONS.

To enable the bureau properly to safeguard itself, so that no unsatisfactory material should be introduced on board ships, a vast system of inspection service was inaugurated. The bureau has representatives at every large manufacturing center, and at every shipyard or large plant where either ships, engine, or steel material is manufactured. These officers and men are more than inspectors—as the bureau's direct representatives they watch all steps taken in the different processes, and are always ready with advice and assistance, to the end that the product may be made more satisfactory and that its cost may be cheapened without loss of quality. In this way they have been of inestimable value to the firms with which they have been associated.

The specifications to which they work are not impossible—they are rigid, but they seek to establish uniformity in the product and to improve the quality of the material, a result always desired by any reputable firm. These specifications, drawn from years of experience and tests, both on shipboard and on shore, seek to obtain the best that can be obtained. For that reason the Federal Specification Board, which now seeks to standardize the specifications under which all material for Government use must be purchased, has adopted extensively the specifications prepared by the Bureau of Engineering of the Navy Department.

But to keep thoroughly abreast of the times and take advantage of the latest development in all branches of engineering, research must be undertaken along those lines which have to do particularly with naval service, and tests made to establish the proper compliance with these specifications. Each navy yard has testing equipment and usually provides a chemist who will analyze those products susceptible of such analysis.

Inspection of rubber, for example, is exceedingly complex, as it is a compound which is capable of being manufactured in many ways, and which necessitates accurate analyses to insure compliance with the specifications. The bureau has constantly aided the manufacturers of these compounds to increase the uniformity of their products, and substantial improvements along these lines have been made. Inspectors of material and machinery make full use of these facilities, with the result that there is little controversy with the manufacturers, who appreciate that these tests are unbiased.

TESTING FINISHED MATERIALS.

Finished products are tested both at the Naval Experiment Station, Annapolis, Md., and at the Testing Laboratory in the New York Navy Yard. A system is in vogue at these places whereby manufacturers are permitted, on application to the bureau and with its approval, to submit samples of their products for exhaustive physical, chemical and life or service tests, upon payment of a small fee to cover the actual cost.

At these stations manufacturers, especially those who have not such facilities themselves, can be assured of a fair test at a minimum of expense. If the test is satisfactory, the manufacturer has the knowledge that his product is up to the standard, and that he can have no hesitancy in marketing it with the assurance that, if it will pass the naval requirements, it should pass any others. If it is not up to standard, it is probable that he will receive such information as will enable him to improve his product.

Many important developments in naval engineering have taken place at these stations, and their full value is hardly appreciated by the average manufacturer of engineering material. There is this stipulation, though, that the results of such tests must not be used for advertising purposes. The Navy must jealously guard the confidence which the manufacturers repose in it, and must preserve at all time the position of an impartial friend and judge.

To supplement these, the naval research laboratory has recently been established. Here it will be possible to take up promising inventions, methods or processes, and develop them into products which will ultimately increase the Navy's efficiency. The horizon of engineering is constantly broadening. An invention or discovery today may later be applied to a great many problems hitherto unsolved.

Emission of ions from heated wires is a case in point. Commencing with the Edison effect, successive inventors have developed the theory of the action and applied it so that today we see the results in Mazda lamps and X-ray tubes, and, which is probably more interesting, in the wonderful realm of radiotelephony. The Bureau of Engineering must constantly be ready to take advantage of such discoveries and see whether or not they are at all applicable to its problems.

KEEPING IN STEP WITH INDUSTRY.

But it is important that naval activities do not run counter to the trend of commercial engineering. Manufacturing in this country has developed enormously and, in certain ways, it has developed by groups; for example, the automobile industry. So great was the demand there for interchangeable parts that the Society of Automotive Engineers, formed from those interested in that work, has established standards so that fittings, bolts, etc., made by different manufacturers, can be used on all types of cars. Standardization of products is greatly to be desired, but there must first come the standardization of the units which compose them. In time of national emergency, when every possible use must be made of all industries, it is essential that the machinist producing similar articles in widely separated

parts of the country use the same allowances, tolerances, etc. That the bureau's designs may be capable of country-wide manufacture, it is essential that they conform to the national standards of design. Hence, the bureau maintains representatives on all such committees of national importance, and it is hoped that this will eventually result in a complete uniformity of manufacture.

A machine which has found great use for commercial purposes is the Waterbury hydraulic speed gear, developed by an employee of the Bureau of Ordnance for use in connection with training and elevating guns. For several years speed gears were used entirely for this purpose and have been developed to a point where they may well be considered at the highest point of perfection in machine design and manufacture. After being successfully used in connection with ordnance material for several years, they finally became of use commercially, and at present are used for steering gears, punch presses, riveting machines, locomotives, textile printing presses and motion-picture machines. While yet in the development state, speed gears show great promise of future use and will undoubtedly be utilized for a great variety of purposes.

METALS AND ALLOYS.

As the proper use of metals is the backbone of the engineering of today, it is of interest to note the bureau's activities in the advancement of this science. As was mentioned before, when the first steam vessels were ordered to be constructed of steel of domestic manufacture it was found impossible to do this immediately, due to the lack of proper knowledge and equipment. The steels that were produced were full of impurities and flaws, and it took long and careful investigations to determine how to remedy this trouble. New methods of production were evolved, improved methods of pouring and heat treating have been developed, and the quality of the production has steadily improved.

The bureau has a membership in the American Society for Testing Materials, and is constantly contributing material of value along these lines. The subject of proper heat treatment of metals goes hand in hand with proper metallographical examination and recording of the tested specimens. The Bureau of Engineering early recognized the great value of proper heat treatment, and has established clauses in its specifications requiring metallographical examination of certain materials to determine whether this has been correctly done. It has established at the Naval Experiment Station a complete set of apparatus for conducting such examinations, and the results obtained from this source have been of inestimable importance in determining the causes of failures and in suggesting remedies. It keeps constantly in touch with the activities of the American Society for Steel Treating, with the object of standardizing the classification in treatment of steels.

Progress in the art of metallography has resulted in the discovery of many new alloys, notably in the steels, where nickel, vanadium and chromium have been introduced, with the idea of toughening or lightening the steels and making them less liable to failure from "fatigue." Whenever possible the Bureau of Engineering has availed itself of the new alloys, in an effort to reduce the weight and space per horsepower of the machinery installations on board ship. This permits more fuel to be carried, with a consequent increase of radius of action, a point greatly to be desired.

Extensive tests and experiments on non-ferrous alloys have been carried out at the New York Navy Yard and the results of a great number of tests have indicated lines which it has been found advisable to follow up. Manufacturers are constantly being called into consultation with regard

to the improvement of their product, as a result of these tests. There is still a great field to be covered in this line, and the Bureau is using every opportunity to find metals of superior quality which can be substituted for those now in use. Considerable progress has already been made in the production of corrosion-resisting aluminum alloys.—"The Iron Age," March 25–April 8, 1926.

TESTS ON FIRST OF THE LARGE DIESELS FOR SHIPPING BOARD.

WORTHINGTON DOUBLE-ACTING TWO-STROKE CYCLE DIESEL DEVELOPED OVER 2900 HORSEPOWER FOR 67 DAYS—FUEL CONSUMPTION 0.462 POUND PER BRAKE HORSEPOWER-HOUR, USING A HEAVY FUEL OIL—ENGINE STARTED ON 85-POUND AIR.

On March 6 the Shipping Board inspectors finished the official tests on the first of the large Diesel engines contracted for by the board, this being a 2900-horsepower double-acting two-cycle engine designed and built by the Worthington Pump & Machinery Corporation. Dismantling of the engine is now in progress and it is expected that the Shipping Board will install the unit in the *Seminole*.

In embarking upon the extensive plans of changing steam-propelled cargo vessels to Diesel drive the board determined that inasmuch as this step was considered a radical one by many in marine circles, each contract for engines should embrace every feature that would guarantee that the units after entering into active maritime duty would prove economical in both fuel and lubricating oil consumption and at the same time so reliable that maintenance would be reduced to a minimum.

In accordance with this desire to safeguard public funds the board required that the engines undergo the following tests:

1. A 30-day continuous run at full load of 2900 horsepower at 95 R.P.M., subject to a tolerance of 1 per cent on power and an overload variation of 3 R.P.M.
2. A six-hour run at 10 per cent overload at 95 R.P.M.
3. A run of four hours with an increase of 10 per cent in the mean effective pressures and with 5 per cent increase in speed.
4. A six-hour run at three-quarters load.
5. Four hours' operating at one-half load.
6. Two hours at one-quarter load.
7. One hour at full load astern.
8. One hour of maneuvering tests.

The Worthington Diesel is a four-cylinder unit, the cylinder bore being 28 inches, with a piston stroke of 40 inches. The outstanding feature is the use of two steel, dome-shaped forgings that go to make up the two ends of each cylinder. Into these steel shells are placed thin cast-iron liners to take the wear due to the piston contact. This four-cylinder unit is a development of the single-cylinder engine completed about two years ago.

After the first of the engines ordered by the Shipping Board had been completed and erected on the test floor at the Buffalo works, arrangements were made to put it through the rigorous tests requried by the terms of the contract. To provide a load, a 25-cycle three-phase alternating-current generator was coupled to the engine shaft and the 30-day continuous run started on February 2. This test was finished on March 4, and without adjustment or examination of any of the parts, the 10 per cent overload was immediately thrown on. Upon the completion of this six-hour run, the engine was successively submitted to the other tests listed. The total period required for these runs was slightly more than 31 days of continuous

operation. The data covering the 30-day run at full load appear in Table I. The engine was shut down at 5 P.M. on March 5 and left under observation but without adjustment or examination of any kind until 9:30 A.M., March 6, at which time the maneuvering tests were begun. Although the machine had stood idle for sixteen hours in the cold building, as soon as the control lever was set to the starting position the engine began firing at the first revolution, even though the fuel pumps had not been primed. The maneuvering tests, which by contract were to be for one hour only, were continued for several hours. These tests, in conjunction with the preliminary shop tests, made a total of 67 days of continuous operation.

Since maneuvering ability is a prime requirement in a marine engine, the results of these tests are of more than passing moment. It was found that the time required for reversal averaged 8.55 seconds, with a minimum of but 4.8 seconds. This time is that elapsing from the moment when, with the engine at full speed in one direction, the starting lever was thrown to reverse, until combustion ensued with the engine rotating in the opposite direction. This rapidity of reversal was not due to any unusual skill of the test engineer, for the operations were performed by a number of the observers, many of whom had had no previous experience with Diesel engines. During the maneuvering twenty-three reversals were obtained with a starting tank capacity of 855 cubic feet, the pressure gradually dropping from 337 to 195 pounds. Additional reversals were made until the pressure dropped to 85 pounds, the total number of reversals being forty-two. This was equivalent to the use of 350 cubic feet of free air per reversal, less than one-half the guarantee figure.

FUEL CONSUMPTION.

It will be seen from Table I that the average fuel consumption for the 30-day run was 0.462 pound per brake horsepower-hour. This was obtained with fuel oil having the following general characteristics of the Bunker "B" specifications:

Gravity, Baumé, at 60 degrees F.	20.19
Flash point, degree F.	177
Burning point, degree F.	270
Viscosity, Saybolt, at 122 degrees F.	181
Hydrogen, per cent.	13.02
Oxygen, per cent.	0.37
Carbon, per cent.	85.29
Water, per cent.	0.11
Asphalt, hard, per cent.	5.95
Sulphur, per cent.	0.37
Ash, per cent.	0.08
Heat value, B.T.U. per pound.	18,733

However, the fuel received showed some variations; some was decidedly inferior, being as low as 14 degrees Baumé. The oil was not centrifuged and as a result grit was constantly working through the fuel pump and spray valves, although no ill effects resulted. The value given for the fuel consumption is subject to a correction which, it is estimated, will bring it down to approximately 0.44 pound per brake horsepower-hour. The generator was rated at 1400 KW. and the efficiency used in determining the load on the engine was 95.1, exclusive of the excitation, this being that determined by an A.I.E.E. code test conducted by the electrical builder. However, there is no doubt that at the load carried, 2900 horsepower, the generator efficiency is lower than at full load, and if so, the actual engine load was higher than computed and the fuel consumption lower. Upon testing the second of the Shipping Board engines, the Worthington engineers

will make an actual input-output test of the generator in order to arrive at the true engine performance. It is probable that the engine in its overload test actually exceeded a load of 3500 horsepower and that the fuel consumption at full load was not more than 0.44 pound per brake horsepower.

Since the engine was engaged in a long-run test, no special pains were taken to reduce the lubricating oil consumption to a minimum. Still, as shown by Table I, the brake horsepower-hours per gallon of lubricating oil were 4104.

TABLE I.—30-DAY TEST ON WORTHINGTON TWO-CYCLE, DOUBLE-ACTING DIESEL.

Date of test	Februray 2 to March 4, 1926
Location	Buffalo, N. Y.
Builder	Worthington Pump & Machine Corp.
Test conducted by	U. S. Shipping Board
Duration of test, hours	720
Revolutions per minute, average	96.4
Total R.P.M.	4,164,952
Brake horsepower, average	2,907
Indicated horsepower, top of cylinder	558
Mean indicated pressure, pound per square inch	93.2
Indicated horsepower, bottom of cylinder	450
Mean indicated pressure, pound per square inch	83.7
Indicated horsepower, total	4,035
Mean indicated pressure, average	88.5
Mechanical efficiency, per cent	72.7
Air compressor indicated horsepower	187.8
Scavenging pump indicated horsepower	194
Fuel per I.H.P.-hr. pound	0.332
Fuel per B.H.P.-hr, pound	0.462
Thermal efficiency, per cent	29.6
Lubricating oil, gallons per 24 hours:	
Power cylinders	2.79
Stuffing boxes	4.25
Air compressor	0.46
Scavenging pump	0.17
Hand feeds	0.26
Crank case	7.07
Total, gallons per day	15.00
Cooling water, gallons per B.H.P.-hr	5.5
Temperatures:	
Cooling water, degrees F. to compressor	35
from compressor	40
Cooling water, degrees F. to jackets	40
from jackets	120
Cooling water, degrees F. to pistons	40
from pistons	105
Exhaust temperature, degrees F.	532
Lubricating oil, degrees F. to engine	95
from engine	96
Fuel oil, degrees F.	89
Engine room, degrees F.	68
Outside air, degrees F.	29

Compressor air temperatures:	
1 stage, degrees F. to cooler	142
from cooler	43
2 stage, degrees F. to cooler	139
from cooler	43
3 stage, degrees F. to cooler	138
from cooler	42
4 stage, degrees F. to cooler	234
from cooler	77
Pressures:	
Injection air, pounds per square inch	890
Standby air receiver, pounds per square inch	869
3 stage of compressor, pounds per square inch	246
2 stage of compressor, pounds per square inch	74
1 stage of compressor, pounds per square inch	24
Scavenging air, pounds per square inch	1 11/16
Lubricating oil, pounds per square inch	12
Cooling water, pounds per square inch	30
Barometer, inch	29.09

Interest, of course, was displayed in the probable cylinder wear. After the completion of the tests the top cylinders and pistons were removed to permit examination of liners, pistons, rings, stuffing boxes, etc. It was found that in a total of 67 days of operation, the wear in the liners averaged 0.002 inch. There was no evidence of scoring in any of the liners or pistons, the surfaces having a fine glaze. Every piston ring was free in its groove and gumming was entirely absent.

NO TROUBLE WITH STUFFING BOXES.

Much discussion has been centered on the question of stuffing boxes of double-acting engines. The stuffing boxes of this engine are provided with metallic packing rings, and no evidence of wear or overheating was present. The piston end is water cooled and the packing rings are exposed to no high temperatures.

In fact, by reason of the design of the piston and lower head, the stuffing-box rings are never in contact with any part of the rod previously exposed to the flames of combustion. From all appearances, the doubt often expressed as to the trouble to be expected with rod stuffing boxes can now be set at rest.

The wear on the running gear, including main journals and crankpins, was not ascertained, although inspection revealed that the wear was insignificant.—"Power," 6 April, 1926.

4400 H.P. DOUBLE-ACTING TWO-STROKE M.A.N. DIESEL ENGINE.

It is hardly too much to say that if marine Diesel engines are to be built appreciably more powerful than those already in existence, development must be along the line of double-acting engines operating on the two-stroke cycle. The advantages of engines embodying these features, as compared with single-acting engines on either the two or four-stroke cycle, are sufficiently obvious. But the difficulties associated with their development are too well known to require emphasis. The type may now be regarded as fully established for units developing shaft horsepowers up to 7000, and as a result of the experience now being gained with such engines in actual use,

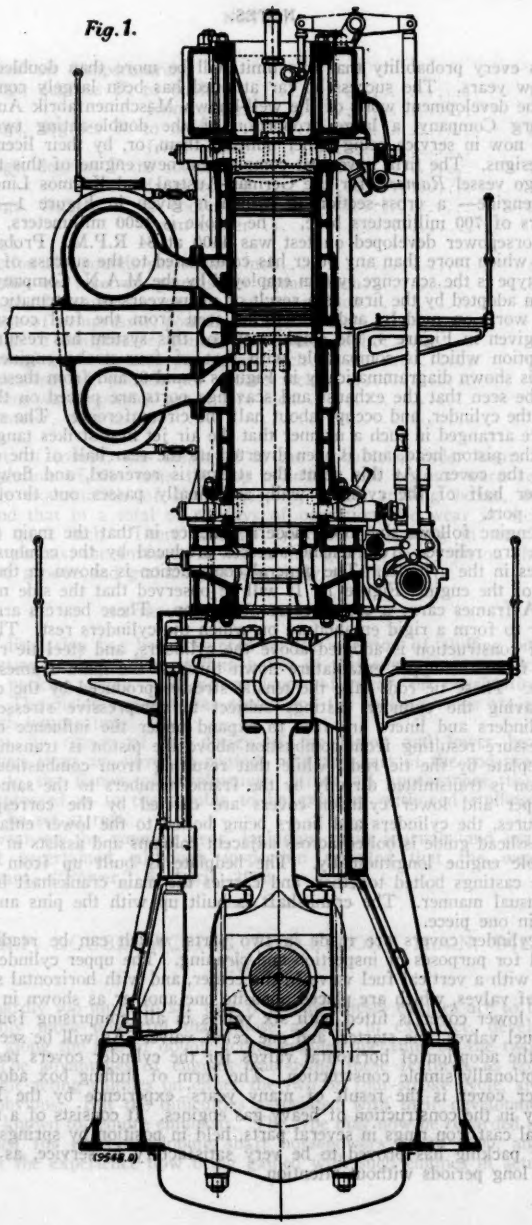
there is every probability that this limit will be more than doubled in the next few years. The success so far attained has been largely contributed to by the development work of the well-known Maschinenfabrik Augsburg-Nurnburg Company, a large proportion of the double-acting two-stroke engines now in service being either built by them, or, by their licensees, to their designs. The firm have just completed a new engine of this type for the cargo vessel *Ramses,* for the German-Austral and Kosmos Line.

The engine— a cross-section of which is given in Figure 1—has six cylinders of 700 millimeters bore. The stroke is 1200 millimeters, and the shaft horsepower developed on test was 4400 at 84 R.P.M. Probably the feature which more than any other has contributed to the success of engines of this type is the scavenge system employed by the M.A.N. Company. This has been adopted by the firm as a result of many years of systematic experimental work on models, and as will be seen from the fuel consumption curves given in Figure 4, the employment of this system has resulted in a consumption which is comparable with that of four-stroke engines. The system is shown diagrammatically in Figures 2 and 3, and from these figures it will be seen that the exhaust and scavenge ports are placed on the same side of the cylinder, and occupy about half the circumference. The scavenge ports are arranged in such a manner that the air jet first strikes tangentially across the piston head, and is then diverted up the rear half of the cylinder wall to the cover. At this point the stream is reversed, and flows down the other half of the cylinder wall, and finally passes out through the exhaust port.

The engine follows the most modern practice in that the main cylinder castings are relieved from tensile stresses produced by the combustion of the gases in the cylinder. The general construction is shown in the cross-section of the engine, Figure 1. It will be observed that the side members of the A frames carry a cross bearer at the top. These bearers are bolted together to form a rigid entablature on which the cylinders rest. The same form of construction is adopted above the cylinders, and steel tie rods are carried from the upper entablature down through the main frames to the bedplate. These tie rods take the tensile stresses produced by the combustion, leaving the cylinder castings subject to compressive stresses only. The cylinders and liners are free to expand under the influence of heat. The pressure resulting from combustion above the piston is transmitted to the baseplate by the tie rods, while that resulting from combustion below the piston is transmitted directly by the frame members to the same point. The upper and lower cylinder covers are carried by the corresponding entablatures, the cylinders and liners being bolted to the lower entablature. The crosshead guide is bolted across adjacent columns and assists in bracing the whole engine longitudinally. The bedplate is built up from several separate castings bolted together, and carries the main crankshaft bearings in the usual manner. The crankshaft is built up with the pins and webs forged in one piece.

The cylinder covers are made in two parts, which can be readily dismounted for purposes of inspection and cleaning. The upper cylinder cover is fitted with a vertical fuel valve in the center, and with horizontal starting and relief valves, which are placed opposite one another as shown in Figure 1. The lower cover is fitted with six valves in all, comprising four horizontal fuel valves, one starter, and one relief valve. As will be seen from Fig. 1, the adoption of horizontal valves for the cylinder covers results in an exceptionally simple construction. The form of stuffing box adopted in the lower cover is the result of many years' experience by the M.A.N. Company in the construction of heavy gas engines. It consists of a number of special cast iron rings in several parts, held in position by springs. This form of packing has proved to be very satisfactory in service, as it will run for long periods without attention.

Fig. 1.



The general arrangement of the valve gear will be clear from Figure 1. The camshaft is situated at the level of the lower transverse bearer above the main engine columns, and is driven by bevels from a counter-shaft. It carries the cams for the fuel and starter valves, and also for the fuel pumps. To reverse the engine the camshaft is moved longitudinally, separate cams being provided for the ahead and astern positions. The starter valves are controlled by an air buffer, which prevents them from opening when the pressure in the cylinder is higher than in the starter pipe. As soon as the engine fires, the admission of starting air ceases automatically. The axial movement of the camshaft is effected by a compressed air cylinder operating through a quadrant, the air being obtained from the starting air reservoir. There is a separate fuel pump for each cylinder, mounted above the camshaft, and operated by cams on the latter as already mentioned. The pumps are arranged so that the fuel delivery coincides approximately with the period of opening of the spray valve, with the result that the

Fig. 2.

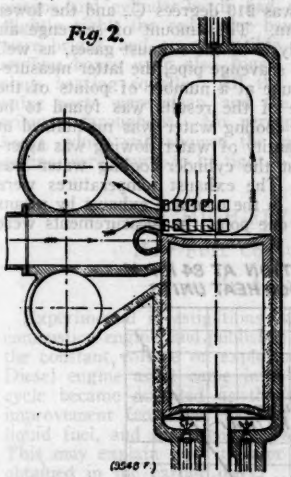
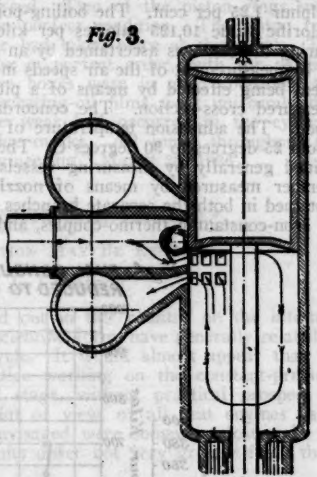


Fig. 3.



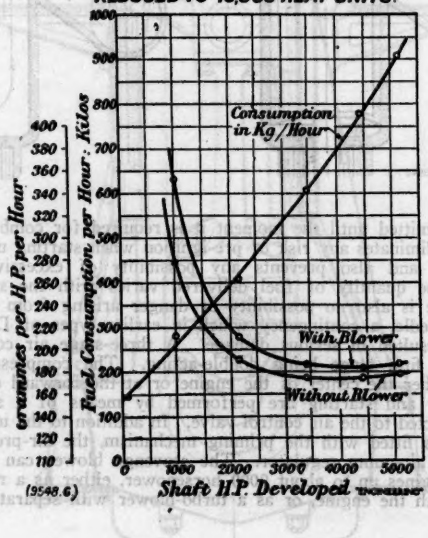
fuel is not admitted until the moment it is required for combustion. The arrangement eliminates any risk of pre-ignition when starting up, or during slow running, and also prevents any possibility of excessive explosion pressures. The quantity of fuel delivered varies with the speed of the engine. There is also no possibility of danger arising from the sticking of the fuel needle, a contingency which in earlier types of Diesel engine occasionally resulted in serious damage. A three-stage air compressor is employed, the first stage being double-acting. The compressor may be mounted at either the center of the engine or at the forward end.

Maneuvering and starting are performed by means of a single lever, which is connected to the air control valve. In addition to the usual gauges, the platform is fitted with the priming mechanism, the air-pressure regulator, and the air-pump regulator. The scavenge blower can be supplied suitable for engines up to about 9000 horsepower, either as a reciprocating set built in with the engine, or as a turbo-blower with separate drive.

The cooling of all parts of the engine has received very careful attention. Forced circulation is employed, the speed of the water at each point being proportioned to the temperature. The pistons are water-cooled, the water being conducted to them through telescopic pipes, which have been so arranged that even if a leak occurs the water can never reach the lubricating oil in the crank-case. Pressure lubrication is employed throughout the engine, with the exception of some small pins and joints. The lubricating-oil pump is of the gear-wheel type, and the usual filters and oil cooler are included in the circuit.

A very complete series of tests was made on the engine after completion at the Augsburg works in January last. The load was provided by three direct-current dynamos. The fuel consumed was checked by weighing. The fuel employed was No. 1 Diesel oil supplied by the German-American Petroleum Company of Hamburg. The oil had a specific gravity of 0.881 at 15 degrees C., and a flashpoint in open crucible of 99 degrees C. An analysis gave a carbon content of 86 per cent, hydrogen 12.6 per cent, and sulphur 1.25 per cent. The boiling-point was 210 degrees C., and the lower calorific value 10,125 calories per kilogram. The amount of scavenge air during the test was ascertained by an analysis of the exhaust gases, as well as by measurement of the air speeds in the scavenge pipe, the latter measurement being effected by means of a pitot-tube at a number of points of the measured cross-section. The concordance of the results was found to be good. The admission temperature of the cooling water was maintained at about 25 degrees to 30 degrees C. The quantity of water flowing was ascertained generally by measuring vessels, but the cylinder-cooling water was further measured by means of nozzles. The exhaust temperatures were obtained in both the separate branches and in the common exhaust by means of iron-constantan thermo-couples, and at one point the measurements were

Fig 4. **FUEL CONSUMPTION AT 84 R.P.M. REDUCED TO 10,000 HEAT UNITS.**



(9548.6.)

checked with the nitrogen-mercury thermometer. The exhaust gases were analyzed with an Orsat apparatus, the samples being taken from three different points in the exhaust pipe.

Curves of fuel consumption taken during the tests are given in Figure 4, allowance being made in the upper curve for the power absorbed by the scavenge pump. It will be seen that when the engine was developing 4460 shaft horsepower, at 84 R.P.M., the consumption was 182.8 grammes per horsepower. The exhaust temperatures measured at the ports and in the pipe were practically identical, and rose from 115 degrees C., when the engine was developing 1000 shaft horsepower, to 280 degrees C. when the power developed was 5110. During the entire period of the test, which covered 180 working hours, the engine ran quietly and smoothly, no adjustments of any sort being necessary, apart from the regulation of the fuel injection. The combustion was good at all loads, the exhaust being invisible under all conditions of loading. After concluding the trial run, the engine was dismantled, and it was found that only slight carbonization of the pistons and cylinder-heads had occurred. Two of the piston rings were slightly tight, the remainder being in perfect condition. The piston rods were bright, and no perceptible wear had taken place at this point. The stuffing boxes were also found to be in perfect order, with the packing rings bearing equally. All the valve boxes were in good condition, the valve ports were clean, and the lubricating-oil films were uniformly distributed throughout. No perceptible wear had occurred in the cylinder bores, which still showed the machine marks. All joints throughout the engine were found to be tight.—"Engineering," March 26, 1926.

THE HIGH-EFFICIENCY OIL ENGINE.*

How Fuel Consumption May Be Reduced.

By Alan E. L. Chorlton.

Experimental investigations carried out in this country on the internal-combustion engine, and published in scientific form, have generally related to the constant volume or explosion cycle. It would almost appear that the Diesel engine as it came into practice working on the constant-pressure cycle became accepted as the final stage, without practical prospect of improvement from the efficiency point of view, of all heat engines using liquid fuel, and the earlier views envisaged were somewhat lost sight of. This may explain why its later results differ not very greatly from those obtained in the earlier days.

Sir Dugald Clerk and Captain Sankey in the "James Forest" lectures in 1904 and 1925, have dealt with the object of further economies in the engine. The former has also many times reviewed the possibilities of well-known cycles and their efficiencies, showing, as is well known, that better results can be obtained by the constant-volume cycle than with that of the constant-pressure cycle.

The practical difficulties in the development of the constant-volume cycle appeared to limit construction to no higher than a certain compression ratio, and hence it followed that the only gains that could be hoped for were those that could be made through other channels, by improvement in the details of the cycle, so to speak.

The main line of development in the improvement of the efficiency of the internal-combustion engine has always been by the increase of the compression or expansion ratio, and it is suggested that it is in this direction that further progress is still possible.

* Extracts from a paper read before the Institution of Mechanical Engineers.

Apart from the apparent belief that practical difficulties beset the use of an increase in the compression ratio with the constant-volume cycle, it is suggested that an advance along these lines may also have been discouraged because of the suppression of heat.

Further, that the higher ratios were impracticable with the explosion cycle for reasons of pre-ignition, etc.; in the main, however, the construction of an engine to work under such conditions of high pressure or heat flow was not a practical proposition.

Later work, however, has shown that there is a distinct practical possibility of progress in the direction of higher ratios, and it is the object of this paper to direct attention to it, also to give some of the results that have been obtained.

Without wishing to repeat what is well known and has so frequently been published, it helps the consideration of the subject to give some generally accepted basic data.

A usual comparison between the two cycles of common practical acceptance is shown by Figure 1 taken from Sir Dugald Clerk's Cantor Lectures, read before the Royal Society of Arts in 1905. This figure shows both C.P. and C.V. diagrams having the same efficiency, but quite clearly shows that the efficiency of the constant pressure cycle depends on expansion to the atmosphere.

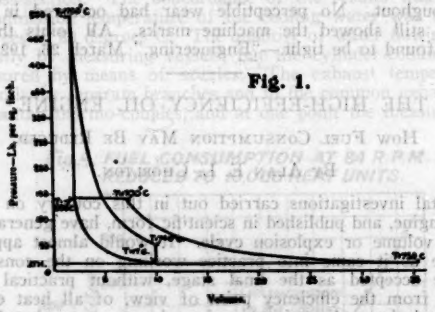


Fig. 1.

An extension of the previous diagram is given in Figure 2 and illustrates a compression ratio of 16 to 1, based on the constant pressure figure, viz., with a maximum temperature of 1700 degrees C as taken by Sir Dugald Clerk for the diagrams shown in Figure 1; the air standard efficiency with this compression ratio is 67.7 per cent against 61 per cent for a compression ratio of 10 to 1.

In reviewing the subject of the higher compression ratios, different ratios of compression were examined, with a view to comparing the prospects of improvement, taking for each case a certain maximum pressure.

Taking the case of a 13.25: 1 compression ratio, the thermal efficiency is 64.6 per cent for a maximum pressure of 1000 pounds per square inch, and a maximum temperature of 1500 degrees C. With a lower calorific value of the fuel equal to 18,300 B.T.U.s per pound, the consumption per I.H.P. under these conditions and allowing for no heat losses would be 0.216 pound per I.H.P. per hour.

With an efficiency ratio of 85 per cent the consumption per I.H.P. becomes 0.254; Captain Sankey's figure of 0.296 was calculated for a compression ratio of 13.1 with much lower maximum pressures, and an efficiency ratio of 90 per cent.

In comparison with the consumption figures actually attained in practice at the present time, these figures indicate that there is much to be done before the unavoidable heat losses are brought down to 10–15 per cent. This indicates particularly that atomization should be given much attention.

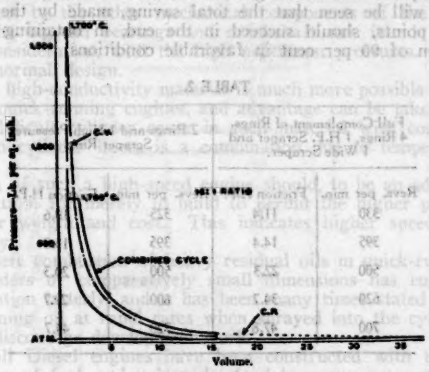


FIG. 2.

With a mechanical efficiency of 85 per cent and an efficiency ratio of 85 per cent, the fuel per B.H.P. for a 13.25 to 1 compression ratio is 0.3 pound, and it should be possible to reach a consumption of this order.

The mechanical losses in I.C. engines are approximately as given in Table 1, from which it is seen that most regard must be paid to the piston rings.

TABLE 1.

Suction .. .	1.4 H.P.	3.4 per cent. of I.H.P.
Piston Friction	2.5 H.P.	6.1 per cent. of I.H.P.
Other Friction (Valve lifting, etc.)..	1.1 H.P.	2.7 per cent. of I.H.P.
Total .. .	5.0 H.P.	12.2 per cent. of I.H.P.

Tests were carried out to determine the ring loss with high pressures, and also to find the effect of variations in piston construction, which may be of interest. These results are given in Table 2.

The normal Diesel engine piston is one with a large number of rings, seven or more, and it can, therefore, from the foregoing, be safely concluded that considerable losses are occasioned thereby. It does appear, therefore, if the mechanical efficiency is to be raised, that the design must follow the line of motorcar or aero practice, that is to say, higher speeds, smaller pistons, fewer rings, etc. Against this, it must not be lost sight of that fluid friction with the higher speed of rubbing of the piston, tends to increase, thus reducing the mechanical efficiency and, therefore, this must also be taken into account.

Gains from savings on other losses obviously cannot be considerable; probably the most important is the use of pressure lubrication throughout, and in connection with this, it will be agreed that, apart from any mechanical gain, it is really an absolute necessity for an engine running at the higher speeds; this the investigation seems to indicate, is the avenue of advance. It will be seen that the total saving, made by the greatest care in all these points, should succeed in the end, in obtaining a mechanical efficiency even of 90 per cent in favorable conditions.

TABLE 2.

Full Complement of Rings. 4 Rings, 1 H.P. Scraper and 1 Wide Scraper.		2 Rings and 1 High Pressure Scraper Ring.	
Revs. per min.	Friction H.P.	Revs. per min.	Friction H.P.
330	11.4	325	10.6
395	14.4	395	13.8
500	22.3	500	20.3
620	34.7	600	29.7
700	47.8	720	44.7

The effect of speed on thermal efficiency is shown by the tests carried out by Sparrow, which indicate that with some engines increasing the compression ratio increases the power loss at low speeds more than at high speeds. The experiments by A. F. Burstall also show conclusively that higher power and efficiency can be obtained from the gas engine by an increase of speed, provided that the engine be suitably designed to run at a high speed.

HIGH PISTON SPEEDS.

It has also been shown by Burstall that it is possible to run efficiently at very high speeds—say, 2000 feet per minute piston speed—with mixtures as weak as 75 per cent of the "correct" without using a stratified charge.

Considering now the possibilities of an engine to meet these conditions, we find that in this country, the development of the engine using simple liquid pressure injection of the fuel direct, without air blast, through an atomizer into the cylinder and working with compression ignition, has recently been more marked.

When applying the high pressures to design, the parts, in the main, which have to be considered are: The outside casing; connecting rods; cylinder covers and bolts; and crankshaft.

By the adoption of cast steel for the main structure, formed as a box of multi-cellular crank casing, adequate provision is made to cover all stresses, even when metal thickness is cut down to ⅜ inch. The connecting rod requires but a small percentage of increase as a strut and the cylinder bolts equally so in tension. Generally, the engine could not be made much lighter except by change of material.

For this type, the average weight per power ratio is but 15 to 20 pounds per B.H.P. In an engine of this design, with cylinder diameters up to 18 inches, a liner of homogeneous hard thin steel allows of great accuracy of manufacture with equality of expansion, and with good cooling closer piston fits are possible, thus securing continued tightness and ability to carry the higher pressures, with freedom from wear, and without any undue material stress under all loads or temperatures.

With a high-pressure lubricating system throughout, engines running at such speeds will give a degree of reliability no less than that of the slow-speed type itself. The piston and rings are subjected to the high pressures for a much shorter time, and working against a hard steel liner the wear is obviated.

With respect to the other mechanical components of the engine, high loading of the big-end bearings and crankshaft are found to be factors worthy of consideration, with the high maximum pressures. They can be covered by normal design.

The use of high-conductivity material is much more possible in the smaller cylinders of quick-running engines, and advantage can be taken to construct the usually difficult cylinder cover in such material with consequent gain. The stress in cylinder liners is a combination due to temperature and to gas pressure.

The design of such a high-speed engine should, to be an advance, include weight reduction, for merely to build to permit the higher pressure would mean greater weight and cost. This indicates higher speeds of rotation as the remedy.

The complete combustion of heavy residual oils in quick-running engines having cylinders of comparatively small dimensions has engaged a good deal of attention latterly, and it has been many times stated that the difficulty of burning oil at rapid rates when sprayed into the cylinders was so great as to discourage development work.

Some small Diesel engines have been constructed with the typical air blast injection of fuel, and it is understood have run successfully, although it is suggested, to reach the high mechanical efficiency indicated, such types could not be considered. Furthermore, their cost of construction tends to be high.

The problem in its other aspect can hardly be brought to quite simple terms, for there are many variations in the way in which it may be attacked. Thus, the form of combustion chamber is important, and there are several different types of such chamber in use (Figure 3).

In the consideration of the comparative merits of these, the question of turbulence plays an important part. In the globular type the connection to the cylinder proper has nearly always been through a conduit of reduced area, and by this means any degree of turbulence can be created. Turbulence goes to improve the degree and speed of burning, and in this wise assists in completing atomization.

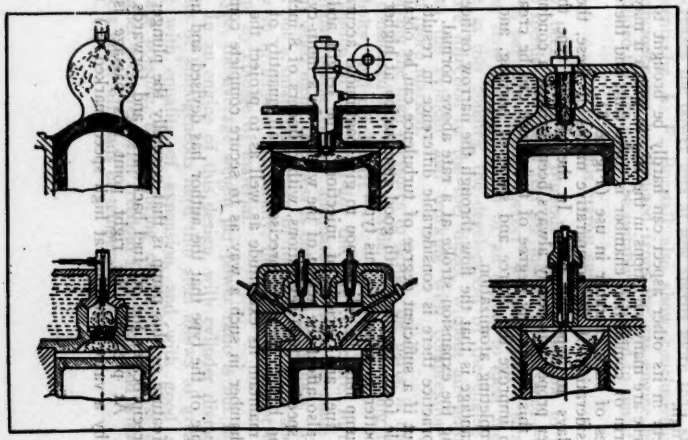
The disadvantage is that the flow through the narrow orifices causes loss of heat during the expansion stroke at a rate above normal.

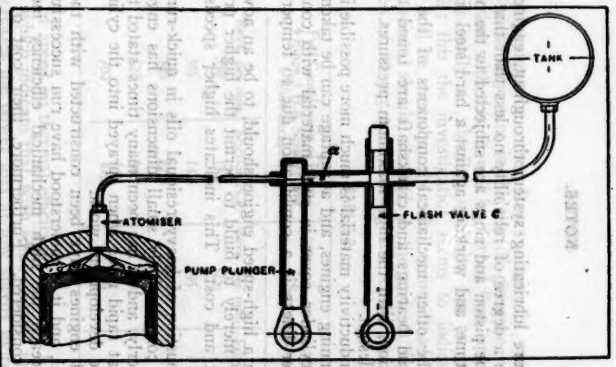
In actual practice there is considerable difference in results between the two forms, but if a sufficient degree of turbulence can be obtained with the flat and simple form, together with good atomization at higher speeds, then it would be better to adhere to this type.

The fuel pump must be relied upon to give continuous accurate measured quantities of fuel and periods of injection. The length and capacity of the fuel pipe also affect the action of the whole. In a multi-cylinder engine running at a speed of 1000 revolutions, with cylinders of 8 inches diameter, the fuel pump has to measure precisely a small quantity of oil for each cylinder and maintain its exactitude as well as to project the fuel into the combustion chamber in such a way as to secure complete combustion in a rapid manner.

The principle of the type that the author has devised and used is shown in Figure 4.

The main feature of this pump is that normally the plunger actuated by an eccentric reciprocates the oil fuel backwards and forwards through the passage (*a*). At precisely the right point this passage is interrupted momentarily by a valve travelling at high speed marked (*c*), the effect of





(Left) Fig. 3.—Showing various shapes of combustion chamber. (Right) Fig. 4.—The injection system of the new Beardmore high-speed Diesel engine.

which is to send a vibration through the column of oil lying between this valve and the atomizer itself, marked (*b*), with a reflex action, when the valve (*c*) reopens the passage (*a*). With this vibration the atomizer-valve is lifted off its seat, and a small amount of oil forced past it at high pressure; it is then drawn on to its seat again by the reflex action, giving an effective cut-off.

TABLE 3.

Test Series.	Duration of Test Hours.	Speeds R.P.M.	B.H.P.	Fuel Lb./ B.H.P./ Hr.	Cooling Water Temperatures.		Fuel.
					Inlet °F.	Outlet °F.	
A	4½	699.3	160.26	0.418	121.5	127.4	Shell-Mex Diesel.
B	3	708.0	172.0	0.385	120.0	128.0	Shell-Mex Diesel.
C	3	707.0	424.0	0.365	140.0	150.0	Shell-Mex Diesel.

Although the mechanical system of injecting fuel has made great progress, it is still, purely from the combustion point of view, not as effective as the air blast, with its excellent turbulence effect. This can be seen by a comparison of consumptions per I.H.P. of the two types. Its great simplicity is the reason for the marked preference that is now shown for it.

On the other hand, the progress made indicates that before long it will equal the air blast on the indicated power, with the consequent increase in lead on that of the brake.

The author concludes with a short description of a quick-running engine manufactured by William Beardmore and Co., which, although not normally in service running at higher pressures, has by reason of its special construction in the use of a material like steel throughout the frame, its higher speeds of revolution, forced lubrication, etc., allowed of tests being made at such higher pressures. These tests have not yet been completed, so the attainment of the highest results has not been reached.

The results of three series of tests are given in Table 3. Series A was carried out on a standard engine with normal maximum pressure by Professor Mellanby. Series B was made on a similar engine with improved

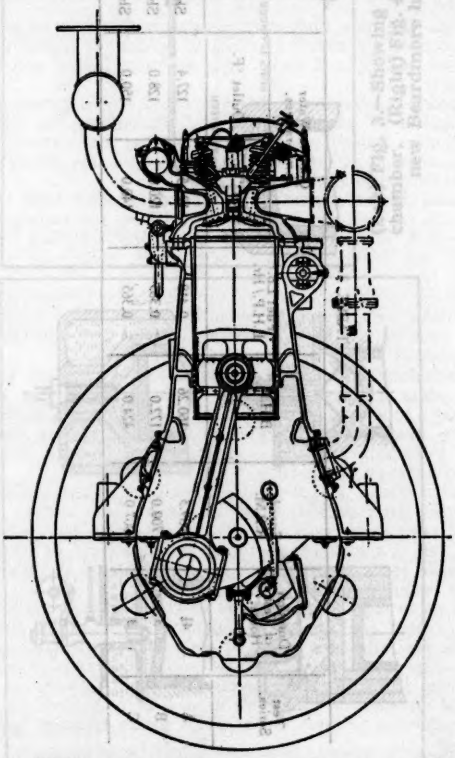


Fig. 5.—Section through the new Beardmore high-speed Diesel engine.

mechanical efficiency, the engine also having a much longer running life. In Series C a high maximum pressure was used. It must be borne in mind when reviewing the results of the last series, that the cylinders were of small diameter with a very high piston speed.

Figure 5 is a section through an 8-cylinder engine, with cylinders 8¼ inches bore by 12 inches stroke, running at 750 R.P.M.

The whole frame is of special cast steel of cellular form, a square cell to each cylinder, the walls continuing to the crankshaft bearings, and carrying directly the stress due to combustion pressure. The liners are thin steel tubes; the pistons and the cylinder covers are of aluminum alloy. This engine is normally rated at 400 B.H.P. and weighs 2¼ tons.—"British Motor Ship," April, 1926.

PISTON TEMPERATURES AND HEAT FLOW IN HIGH-SPEED PETROL ENGINES.*

By Professor A. H. Gibson, D.Sc.

This paper deals in the main with the results of the measurement of piston temperatures made by the author during the past seven years. Except where otherwise stated, temperatures were obtained at different radii in the crown of the engine piston under full load operating conditions, by the method of cooling curves. Such a comparison as is possible between the results obtained by this method and by the use of thermocouples fixed in the piston, indicates that these are in fair agreement.

The first series of tests was carried out on an air-cooled aluminum cylinder, 100 millimeter bore by 140 millimeter stroke, with the object of determining the effect of a variation in: (*a*) the piston clearance; (*b*) the material of the piston; (*c*) the design of the piston; (*d*) the bearing area (*e*) the working mixture; (*f*) the compression ratio; (*g*) the spark advance. All these tests were carried out under full throttle, full load conditions, and except in those tests designed to determine the effect of a variation in the working mixture, the petrol flow was adjusted as nearly as possible so as to give the weakest mixture capable of giving full load. Piston temperatures were measured at the center and at radii of 0.5, 1.0 and 1.5 inch, along a line joining the center and the spare sparking-plug position. This is at the back of the cylinder at the side remote from the cooling blast, and consequently at the hottest side of the cylinder. For this reason the hottest point of the piston is not at the center, as might be expected with walls of uniform temperature, but at a point nearer the hottest wall, and apparently about 0.4 inch from the center. The temperature of the cylinder-wall was measured at a series of points down the back of the cylinder, and from these measurements the mean temperature of the back wall has been obtained. The main results of the tests are given in Table I.

Effect of Clearance.—One of the pistons was first tested with a clearance (cold) of 0.024 inch, was then turned down to a smaller diameter giving a clearance of 0.045 inch, and was retested. The results are shown in lines (1) and (2) of Table I, from which it appears that the effect of the increased clearance is to increase the drop in temperature between the edge of the piston crown and the wall by 29 degrees C., from 27 degrees C. to 56 degrees C. The temperature of the hottest point in the piston is increased by 34 degrees, but as the cylinder wall is 8 degrees hotter the temperature gradient from center to edge of the piston is sensibly the same

* Paper read before the Institution of Mechanical Engineers on Friday, January 22, 1926. Abridged.

in both cases. The results indicate that the clearance should be reduced to a minimum compatible with successful operation, but that any such slight changes in clearance as are usual in good practice do not seriously affect the temperature.

TABLE I.

Ling.	Piston.	Type.	Alloy.	Petrol. Pints per B.H.P. Hour.	B.H.P.	Temperature of Piston, deg. C, at					Mean Back Wall Temp. deg. C.	Difference of Temp. between Piston and Wall.	
						Centre.	0·5 in. Rad.	1·0 in. Rad.	1·5 in. Rad.	Edge			
						Deg.	Deg.	Deg.	Deg.	Deg.	Deg.	Deg.	
1	A	Ribless	B 4	0·504	17·85	194	206	184	175	173	146	27	—
2	A1	As A with increased clearance.	B 4	0·605	17·90	234	248	226	213	210	154	56	
3	B	As A	B 24	0·506	18·00	197	207	184	176	174	148	26	—
4	C	Ditto.	B 22	0·585	18·20	205	219	200	187	183	153	25	
5	D	Heavily ribbed.	B 4	0·585	17·85	210	223	198	182	177	152	25	—
6	E	Ditto.	B 18	0·508	17·70	236	246	215	194	186	160	26	—
7	F	Ditto.	B 25	0·605	17·65	224	236	211	194	188	159	29	
8	G	Lightly ribbed	B 4	0·597	18·10	215	238	218	197	184	156	28	—
9	K	Cast-iron	C 1	0·644	17·05	410	430	387	310	220	156	64	—
10	B1	As B with holes in skirt.	B 24	0·620	17·65	206	218	198	188	185	146	39	—
11	H	Heavily ribbed	B 4	0·605	17·25	244	250	230	208	187	155	32	Magneto Advance 35 deg.
12				0·604	17·50	254	259	238	210	189	156	33	48 deg.
13				0·827	13·00	239	266	249	213	189	156	33	18 deg

Effect of a Variation in Design.—In this series a comparison was made between pistons of the same aluminium alloy (Cu 7 per cent; Sn 1 per cent; Zn 1 per cent) but of different design. For this series three designs were available. The essential details are tabulated below.

The results of these tests are shown in lines 1, 5 and 8 of Table I, from which it appears that the difference between wall and piston skirt is sensibly the same for all the pistons, being respectively 27 degrees, 28 degrees and 25 degrees for A, G and D. Taking the temperature of the hottest point as a criterion, however, the ribless piston, shows to advantage. These temperatures are respectively 206 degrees, 238 degrees, and 223 degrees for A, G and D. Comparing the ribless piston with design D, it appears that the former is slightly lighter, and in spite of its slightly greater clearance in this particular case, was about 20 degrees cooler. It is about 30 degrees cooler than the lighter piston G.

Piston.	Type.	Thickness at Crown.	Mean Diameter.		Weight.
			Lands.	Skirt.	
		Mm.	In.	In.	Lb.
A	Hollow crown, ribless	4·0	3·9085	3·9175	1·408
G	One shallow diametrical rib in each direction	6·5	3·9075	3·9150	1·262
D	Four deep ribs at right angles to gudgeon pin, one shallow rib parallel to gudgeon pin	6·5	3·9125	3·9185	1·425

Effect of Perforations in Piston-Wall.—In order to investigate the effect of reducing the bearing area of the piston, a series of holes was drilled through the walls. Piston B was used for this test. Thirty holes in all were bored—twenty-two of ¾-inch diameter and eight of ½-inch diameter. The total area of these holes is 4 square inches, and as the peripheral contact area of the normal piston is 33.4 square inches, this gave a reduction in area of 12 per cent. The reduction in weight was 0.10 pound. The results of the tests are shown in lines 3 and 10 of Table I. The result of reducing the bearing surface is to increase the drop in temperature between piston crown and wall from 26 degrees to 39 degrees. The temperature of the hottest point of the piston is increased by 11 degrees from 207 degrees to 218 degrees.

Effect of a Variation in the Material of the Piston.—For these tests, pistons of five different aluminium alloys and one cast-iron piston were used. The compositions of the aluminium alloys and their measured conductivities in the chill-cast state at 200 degrees C. are tabulated.

Piston	Designation of Alloy.	Composition of Alloy (approx.).	Conductivity. C.G.S. Units.
A, D	B 4	Al 91, Cu 7, Sn 1, Zn 1 per cent.	0·40
B	B 24	Al 93·5, Cu 4, Mg 0·5, Fe 2 per cent.	0·38
C	B 22	Al 92·5, Ni 1, Mg 6·5 per cent.	—
E	B 18	Al 89·5, Cu 8, Sn 1·5, Mn ½ per cent.	0·27
F	B 25	Al 89, Cu 9, Ni 2 per cent.	0·37
K	Cast-iron	—	0·11

Of these, A, B, and C were cast from the same patterns, and D, E, and F were also identical in design. The results (lines 1, 3, 4, and 5, 6, 7 of Table I) show that there is sensibly no difference in the temperatures of various pistons with the exception of E, which is approximately 20 degrees C. hotter than the remainder, at the center, under full-load conditions. It may be shown that in similar pistons the temperature gradient along a radius should be inversely proportional to the conductivity of the alloy. Comparing pistons D and E, the ratio of the conductivities is: 1.34, while the ratio of the temperature gradients is: 1.30. Evidently the higher temperature in piston E is due to its lower conductivity. For the comparison of cast-iron and aluminium alloy as a piston material, it is unfortunate that pistons of identical design were not available. Two pistons were chosen of the same general type. Each has a shallow strengthening rib running along the line of the gudgeon-pin and a second rib at right angles to this. The thickness of the crown and walls of the aluminium piston, is, however, much greater than that of the cast-iron piston, which is of normal design for a piston of this material. The details are given in the following table:

Piston.	Material.	Thickness of Crown.	Mean Diameter (cold).		Weight.
			Lands.	Skirt.	
		Mm	In.	In.	Lb.
G	Al (B 4)	6·5	3·9075	3·9150	1·262
K	Cast-iron	3·0	3·9130	3·9205	1·776

The results of the comparative tests are shown in lines 8 and 9 of Table I. From these it appears that the maximum temperature is about 200 degrees higher in the cast-iron piston. The difference in temperature between the hottest point and the edge is 210 degrees in the cast-iron and 54 degrees in the aluminium, and the difference of temperature between piston edge and wall is 64 degrees in the cast-iron and 28 degrees in the aluminium piston. It will be noted that the mean clearance (cold) of the cast-iron piston is 0.0055 inch less than that of the aluminium piston. At the mean working temperatures, however, calculation shows that the difference in clearance is only 0.0003 inch, that of the cast-iron piston being the greater. This small difference is not such as to account for any appreciable difference in the temperature drop from the edge of the piston to the wall. While, owing to the steepness of the temperature radius curve in the cast-iron piston, the error in deducing the edge temperature is likely to be greater than in the case of an aluminium piston, some such difference as is here indicated is to be expected, since, with a given temperature at this point, the mean temperature of the piston-skirt will necessarily be lower in a cast-iron than in an aluminium piston, owing to the higher conductivity of the latter material, and, in the case in question, owing to the smaller cross-sectional area of the metal in the skirt of the cast-iron piston. The difference between the power developed with the two pistons is appreciable. The aluminium piston gives 18.10 H.P. as compared with 17.05 H.P. for the cast-iron piston, a difference of 6 per cent, and this with about 8 per cent less petrol per brake-horsepower-hour. If the two pistons were developing the same power, the maximum temperature of the cast-iron piston would be approximately 440 degree C., as against 230 degrees C. for the aluminium piston.

Effect of Compression Ratio.—The results of a series of tests, carried out to determine the effect of varying the compression ratio, are given in Table II. In these tests only the temperature at the center of the crown was measured. From these it appears that, with a maximum load mixture, the temperature diminishes slightly as the compression ratio is increased between the limits 4.6 and 5.3. Also, if the difference of temperature between the center of the piston and the wall be taken, this diminishes from 59 degrees with a 4.6 compression ratio to 44 degrees with a 5.3 ratio, although the brake horsepower is 3 per cent greater with this higher ratio. This indicates that the piston is receiving less heat from the working fluid with the higher compression ratio.

Measurements of cylinder-wall temperatures on a similar cylinder, in each case with the weakest maximum load mixture, gave the following results:—

Compression Ratio.	B.M.E.P.	Petrol pints per B.H.P.–hour.	Mean Temp. of Barrel Deg. C.		Remarks.
			Top.	Bottom.	
			Deg.	Deg	
4·6	116·2	0·586	180	105	
5·0	119·3	0·580	170	95	
5·4	122·0	0·542	157	89	
5·8	125·0	0·525	154	85	
6·2	129·0	0·531	183	110	Occasional pinking
6·4	133·0	0·575	212	135	Heavy pinking

These confirm the piston-temperature measurements in showing that the heat given to the walls and hence to the piston is reduced as the compression ratio is increased, so long as the ratio is not so great as to give rise to detonation of the working fluid. This is due to the increased efficiency of working with the higher compression ratios. A smaller amount of petrol is consumed per minute and a larger proportion of the heat of combustion of this is converted into work, leaving a smaller amount of heat to be lost to the walls and piston, and in the exhaust gases.

Effect of Speed and of Mixture Strength.—Tests 7 and 8 of Table II show that the maximum temperature of the piston is only increased 7 degrees C. by increasing the speed from 1800 R.P.M. to 2000 R.P.M. Moreover, this 7 degrees is to be accounted for by an increase in the wall temperature, so that change in speed over this range has no direct marked effect on the piston temperature. Table II also shows the effect of a variation in the strength of the petrol mixture, on the temperature of the piston, both with cast-iron and aluminium pistons. These indicate that the weakest mixture capable of giving the maximum power output also gives the maximum piston temperature. Experience shows that this also gives the maximum wall temperature. The cooling effect of a very strong mixture is well shown by the figures in lines 5 and 6, and 10 and 11 of Table II.

Effect of Spark Advance.—Experience shows that as the spark advance in a petrol engine is increased, so long as the advance is not so great as to give rise to detonation, the output of the engine is increased; the temperature of the exhaust valve is appreciably reduced; while that of the cylinder head and walls is usually slightly increased. On the whole it

would be anticipated that the effect on the piston temperature will not be very great. The hotter walls will normally be accompanied by a hotter piston, but this effect will be to some extent counterbalanced by the reduction in the heat reaching the piston by radiation from the cooler exhaust valve. The results of tests to investigate this point are given in lines 11 to 13 of Table I. These tests were carried out at 1800 R.P.M. on an aluminium piston having somewhat thinner crown than those previously described. The spark advance in these tests was respectively 18 degrees,

TABLE II (All Tests Carried Out on Same Aluminium Air-Cooled Cylinder).

Test No.	Speed.	B.H.P.	Brake M.E.P.	Petrol Pints per B.H.P.	Compression Ratio.	Temp. of Centre of Piston.	Mean Temp. of back wall of Cylinder.	Material
						Deg. C.	Deg. C.	
1	1,800	17·4	113	0·615	4·6	214	156	Aluminium piston.
2	,,	17·6	114	0·648	,,	216	154	
3	,,	17·4	113	0·700	,,	210	152	
4	1,800	18·4	120	0·575	5·1	216	153	
5	,,	18·2	119	0·620	,,	213	150	
6	,,	17·8	116	0·860	,,	196	141	
7	1,800	17·9	117	0·605	5·3	105	151	
8	2,000	16·9	111	0·595	,,	202	159	
9	1,800	16·5	109	0·643	4·6	356	155	Cast-iron piston.
10	,,	16·6	110	0·675	,,	380	155	
11	,,	16·5	109	0·795	,,	336	141	

35 degrees, and 48 degrees. The results show that, while the effect on piston temperature is not great, the highest temperature was obtained with the smallest advance, whilst the lowest temperature was attained with an advance of 35 degrees.

Tests on a 5-inch Aluminium Piston.—These measurements were made on the aluminium piston of a 5-inch by 5-inch air-cooled cylinder. The tests were carried out to determine the distribution of temperature down the skirt of the piston, and no crown temperatures were taken. The temperatures were taken on the back side of the cylinder under the exhaust port, at about 45 degrees to the axis of the gudgeon pin. Down this line the mean temperature of the cylinder wall is 197 degrees C. The following table shows the mean results of the tests, which were carried out with a compression ratio of 5.0, at a speed of 1700 R.P.M. and with an air temperature of 21 degrees C.

B.M.E.P.	Petrol pts. per B.H.P.-hour.	Temperature, Deg. C. of Piston Skirt at					
		0·55	0·90	1·8	2·6	3·5	Inches from the top of the piston.
124·5	0·637	Deg. 248	Deg. 245	Deg. 237	Deg. 218	Deg. 168	—

From these figures it appears that the temperature at the edge of the head is approximately 250 degrees C., so that the drop at this point across the oil-gap is 53 degrees C. The mean temperature of the skirt at this point was, however, only 222 degrees C., so that the difference between the mean wall temperature and the mean skirt temperature at this part of the wall was 25 degrees C. The clearance of this piston (cold) varied from 0.045-inch over the top lands to 0.035-inch over the bottom lands and 0.025 over the skirt of the piston.

Tests on 5½-inch Slipper-type Piston.—The following tests were carried out on an aluminium piston of the slipper type running in a 5½-inch by 6½-inch steel air-cooled cylinder with a compression ration of 4.6. The temperature of the piston was measured at the center and at a point ¾ inch from the edge, in line with the sparking-plug hole, which is midway between the back and left-hand sides of the cylinder. The mean results of a number of measurements are as shown in Table IIA.

In these tests the air temperature was 19 degrees C. and the spark advance 25 degrees, the maximum possible without detonation. The mean wall temperature is 179 degrees, and as the temperature of the edge of the piston at this point is approximately 239 degrees, the drop across the oil-gap is 60 degrees C. If this is uniform around the periphery, the hottest part of the piston would be at the back of the cylinder, where the edge temperature would be 269 degrees, and the coolest point would be at the opposite end of the diameter where the temperature would be 174 degrees. This indicates that the center of the piston is approximately 24 degrees hotter than the mean of the opposite edges, and about 85 degrees hotter than the mean of the opposite cylinder walls. The thickness of the top of this piston is 8.5 millimeter (0.335-inch). Comparing this piston with the 100-millimeter pistons previously examined, it appears that the temperature drop across the oil-gap is about twice as great, as might be expected from the fact that its bearing surface is relatively considerably

less. The mean temperature drop from the hottest point to the edge is, however, much less. This is probably due partly to the fact that this cylinder could only run on a very rich mixture which in itself tends to coolness in operation, but mainly to the design of the piston, in which

TABLE IIA.

Revs.	B.M.E.P.	Petrol pts. per B.h.p.-hour.	Piston Temperature. Deg. C.		Temperature of Wall at Points shown in Sketch. Deg. C.				
			Centre.	½ in. from edge.	T 1.	T 2.	T 3.	T 4.	
									L.H.
1,450	106·2	0·895 (weakest maximum load mixture)	Degrees. 246	Degrees. 241	Degrees. 190	Degrees. 157	Degrees. 129	Degrees. 117	Back.
					263	219	185	167	
					145	129	101	91	Front.

the gudgeon-pin is carried by two heavy lugs which join the crown at points between the center and the edge. Much of the heat is transferred from the head down these lugs to the connecting-rod. Furthermore,

the piston-crown has a raised lip 11 millimeter deep around its periphery. This will receive a considerable amount of heat from the working fluid which must tend to increase the temperature around the periphery. The maximum wall temperatures attained in any tests on this cylinder were at 1650 R.P.M. on a weak mixture (0.80 pint per brake-horsepower-hour) at 98.0 B.M.E.P. These were 54 degrees higher than those recorded above. No piston temperatures were taken in these tests, but such wall

TABLE III.

Test.	Revs.	B.H.P.	Petrol pts. per B.H.P.-hour.	Cyl.	Temperatures deg. C. of					Temp. deg. C. of		Remarks.
					Piston.		Liner.			Outlet Water.	Oil.	
					Centre.	Edge.	1	2				
					Deg.	Deg.	Deg.	Deg.		Deg.	Deg.	
1	1,880	189·0	0·590	H	240	270	222	178		62	62	One sparking-plug in operation on outside of each cylinder.
				C	190	210	164	122				
2	1,880	187·0	0·592	H	240	220	173	135		58	62	One plug on inside of cylinder H; one on outside of C.
				C	192	212	164	125				
3	1,880	188·0	0·585	H	244	275	226	180		63	62	As in 2, but with one plug on outside of cylinder H.
				C	195	214	166	125				
4	1,900	187·1	0·542	H			170	138		65	65	Plug on inside of cylinders H and C.
				C			145	112				
5	1,920	194·0	0·585	H	235	218	174	152		67	72	Plug on inside of cylinders H and C. Pistons with reduced clearance.
6	1,890	184·5	0·610	H			240	190		64	62	Two plugs in each cylinders.
				C			180	138				

temperatures would correspond to a maximum piston temperature of the order of 300 degrees C. This piston is probably cooled very appreciably by oil-splash from the crank-case.

Tests on Aluminum Pistons in a water-cooled Engine.—These tests were carried out on an eight-cylinder water cooled Vee aero-engine having cylinders 120 millimeter (4.72-inches) bore by 180 millimeter (7.09 inches) stroke, and a compression ratio of 5.06. Two of the cylinders of this engine had previously given trouble from burnt pistons. A preliminary investigation in which temperatures of the cylinder liner were measured by thermo-couples showed that the liners in these two were considerably hotter than those in the other cylinders, the difference at a point 1 inch below the level of the piston-crown, at the inner dead center being as much as 66 degrees C. under normal running conditions. Further investigation was then confined to the hottest and coolest cylinders, denoted respectively by H and C. Temperatures were obtained by the cooling-curve method at the center of the piston and at a point ¼ inch from the edge in line with the sparking-plug hole at the outside of the cylinder. The results of typical tests are shown in Table III. The liner temperatures shown in the Table were taken on the outside of the cylinder in line with the thermo-couple hole near the edge of the piston at which the piston temperature was measured, and respectively 1.0 inch and 3.3 inches below the level of the crown of the piston at the inner dead-center.

The figures show that the temperature at the edge of the piston in cylinder H is about 270 degrees C. when the sparking-plug is at the outside of the cylinder (that is, adjacent to this side of the piston) and about 220 degrees C. when the plug is at the opposite side of the cylinder. This change in the position of the sparking-plug produces a change in the temperature of the top of the liner in this cylinder, from 222 degrees to 173 degrees, so that the difference in temperature between wall and piston at this point is sensibly constant at 48 degrees. At point (2) the difference varies from 92 degrees with an outside plug to 85 degrees with an inside plug. It will be noted that the temperature at the center of the piston is in each case lower than at the edge under the sparking-plug. Comparing tests (2) and (3), in which the only difference was the changing of the plug position in cylinder H, it appears that the temperature of the piston-center is practically the same in each, having a mean value of 242 degrees, but that the edge temperature changes from 220 degrees to 275 degrees. This gives a mean edge temperature of 247 degrees, which indicates that with a uniformly hot liner the temperature would be sensibly constant over the piston crown. This would tend to show that the piston-cooling is very largely due to oil-splash, which is known to be excessive in this particular engine.

The clearance in the two cylinders was measured after the tests. They are as follows:—

Diametrical Clearance.	Piston H.	Piston C.
	Inch.	Inch.
Mean at top land	0·060	0·063
Mean over skirt	0·017	0·0185

The clearance at the top land appears to be large compared with that usually allowed in other water-cooled engines, and after running some time a solid layer of carbon was deposited over this land. As this is a bad thermal conductor, it was thought possible that a piston having a reduced clearance at this point might run cooler. To test this, two pistons were

made having the clearance over the top land reduced to 0.025 inch. These were put into cylinders H and C, and measurements, the mean of which are shown in test No. 4, were made on the cylinder H. A comparison of tests Nos. 1, 2, 4 and 5, would tend to show that this alteration is beneficial. Comparing tests Nos. 4 and 5, in which the plug positions were identical, it appears that in spite of the fact that the power was 7 H.P. greater in test (5), the liner temperatures at point (1) were sensibly the same as in test (4). On the other hand, the temperatures at point (2), lower down the liner, were appreciably greater in test (5), which would rather indicate that in this test the piston was getting rid of more heat to the liner in its downward travel.

Again, comparing tests Nos. 2 and 5, although the power was greater in No. 5, and the oil and cooling water were each about 10 degrees hotter than in test No. 2, the temperature of the piston in cylinder H was slightly less in test No. 5 than in No. 2. The difference of temperature between the edge of the piston and the liner at point (2) was reduced from 85 degrees in test No. 2 to 66 degrees in No. 5. It will be noted that in this test, as in No. 2, the center of the piston is hotter than the edge, which in both cases is diametrically opposite to the operating sparking-plug. The results of the tests indicate that under normal conditions of operation, using two sparking-plugs, the mean edge temperature in cylinder H is about 265 degrees. The mean temperature of the working portion of the liner probably does not differ greatly from that at point (2), or about 190 degrees C. This gives a drop of 75 degrees C. across the oil film at the edge of the piston-crown. This was obtained with a piston clearance over the upper land, of 0.060 inch, and compares with the values of 27 degrees obtained with a clearance of 0.024 inch, and 56 degrees with a clearance of 0.045 inch, in a 100 millimeter piston, and with the value of 53 degrees obtained with a clearance of 0.045 inch in the 5-inch piston dealt with earlier. An interesting feature of these tests is the comparatively large difference between the temperature of the cylinder liner and that of the outlet jacket water. With two sparking plugs in operation, the difference between the temperature of the liner at point (1) and that of the cooling water, is 180 degrees C. in cylinder H, and 120 degrees in cylinder C. At point (2) the differences are 130 degrees C. and 80 degrees C.

Cyclical Fluctuations of Temperature in the Surface of a Piston.—The figures already given denote the mean temperatures to be anticipated in pistons of the types described, and under the conditions of operation outlined. The question, however, arises as to whether, due to cyclical fluctuations of temperature in the metal, the maximum skin temperature may be appreciably in excess of the mean temperature. The suggestion that this is the case has, in the past, been put forward to account for the burning of aluminium pistons in aero-engines, under operating conditions in which their maximum temperatures were apparently much below the eutectic melting point of the alloy. On the assumption that the working fluid in contact with the metal has a cyclical fluctuation of temperature of $\pm$ A degree C.; that the heat transmission from the gas to the metal is directly proportional to their difference of temperature; and that this produces a simple harmonic fluctuation of temperature in the metal, it may be shown that in the case of a large flat surface, the magnitude "a" of the semi-amplitude of the fluctuation of surface temperature is given by—

$$a = \frac{A e'}{\sqrt{\pi n \rho s k + (e' \sqrt{\pi n \rho s k})^2}} \quad \ldots\ldots (1)$$

where n is the number of explosions per second;
ρ is the density of the metal;
s is the specific heat of the metal;
k is the thermal conductivity of the metal;
e' is the receptivity, that is, the heat transmitted from the gas to the surface per second per unit area, per degree difference of temperature between metal and gas.

If e' is relatively small, as is shown later to be the case—

$$a = \frac{A\,e'}{\sqrt{2\pi n \rho s k}} \text{ approximately} \quad \ldots\ldots (2)$$

So that $\frac{a}{A} \propto \frac{1}{\sqrt{n}}$ for a piston of given material; $\frac{a}{A} \propto \frac{1}{\sqrt{\rho s k}}$

for pistons of different materials operating at the same speeds of rotation. The assumption that e' in the above formulæ is a constant is, however, very far from being true. Actually, Sir Dugald Clerk's experiments on the rate of cooling indicate that under the conditions obtaining in a gas-engine, the heat transmission "h" per unit area per second, is proportional to ϕ^n, where ϕ is the difference of temperature between gas and metal and where n is very nearly equal to 2.* If n be assumed equal to 2 for convenience in calculation, the results, though not quite accurate, will much more nearly represent the true state of affairs than will those deduced on the assumption that $n = 1$, and if $n = 2$ it may be shown that the formula corresponding to (2) is—

$$a = \frac{e\left\{(T_1 - \theta_0)^2 - (T_2 - \theta_0)^2\right\}}{2\sqrt{2\pi n \rho s k}} \quad \ldots\ldots (3)$$

where T_1 is the maximum temperature of the gas;
T_2 is the minimum temperature of the gas;
θ_0 is the mean temperature of the surface;
$e\phi^2 = h$ = heat transmitted per unit area per second, from gas to surface.

Value of e.—A knowledge of the value of e would be of great value, not only to enable the magnitude of the surface fluctuation to be calculated, but also to enable the rate of heat flow into the exposed surface of an internal combustion engine to be determined. Its value depends on the temperature and density of the hot gases, and on their velocity over the metal surface. Its direct experimental determination is one of much difficulty, and no definite values have hitherto been available, especially at the temperatures and with the degree of turbulence which obtains in a high speed petrol engine in operation. A fair approximation to its value under such conditions may, however, be deduced from measurements of piston temperatures, for if the temperature gradient across the face of a piston is known, the rate of reception of heat h per unit area of the face, or rather the difference between the rate at which heat is being received by the working face and dissipated at the lower face of the piston crown, is given very approximately by—†

* Clerk, "The Petrol and Gas Engine," vol. i, page 204 (1909), page 247.
† Proc. Inst. C. E., vol. clxxvi, 1908-9.

$$h = \frac{4\,k\,t\,\theta}{a^2 + t^2},$$

where θ is the difference of temperature between the center and a point at radius a;
t is the thickness of the piston crown;
k is the conductivity of the material.

If $(\theta_1 - \theta_2)$ be the differences of temperature at two radii a_1 and a_2, this becomes—

$$h = \frac{4\,k\,t\,(\theta_1 - \theta_2)}{(a_2^2 - a_1^2)}.$$

In a piston in which the thickness of the crown varies with the radius, it, over any portion of the crown between radii a_1 and a_2 the thickness is proportional to the radius, so that $t = Ca$, we have—

$$\delta\theta = \frac{h\,(a_2^2 - a_1^2)}{4\,k\,t}$$

$$= h \times \frac{2\,a\,\delta a}{4\,k\,t}$$

$$= h\,\frac{\delta a}{2\,k\,C}.$$

$$\therefore \theta_1 - \theta_2 = \frac{h}{2\,k\,C}\int_{a_1}^{a_2} da$$

$$= \frac{h\,(a_2 - a_1)}{2\,k\,C}.$$

In a number of the author's pistons, namely, A, A′, B and C, Table I this condition is satisfied over the range of radii 0.5 inch to 1.0 inch (1.27 centimeter to 2.54 centimeter), the value of C. being 0.31. A further idea of the value of e may be obtained from such measurements as are available for the fluctuations of temperature in the metal of the combustion space and piston of internal-combustion engines,* and also from Sir Dugald Clerk's cooling curves for gas or engine cylinders.† An examination by the author of the data available from these sources, and from the temperature gradient measurements of the present paper, and of those by other observers,‡ leads to the conclusion that the value of e, for use in equation (3), increases with the turbulence, and under normal conditions of operation, and with the compression ratios usual in practice, varies from 3.6×10^{-6} C.G.S. units in a gas engine with a cylinder of 6 inches to 12 inches diameter at 200 R.P.M., to about 11.0×10^{-6} in a high-speed petrol engine at 1800 R.P.M. Adopting the latter value and assuming $T_1 = 2400$ degrees C., $T_2 = 250$ degrees C.; $\theta_0 = 250$ degrees C. in a petrol engine at 1800 R.P.M. having an aluminum piston for which $\rho = 2.9$; $s = 0.21$; $k = 0.38$—

* Coker and Scoble. Proc. Inst. C. E., vol. cxcvi, 1913-14, Part II.
† Proc. Roy. Society. A, vol. lxxvii, 1906, page 499.
‡ Phil. Mag., vol. xlvii, May, 1924, page 883.

$$\frac{e\left\{(T_1-\theta_0)^2-(T_2-\theta_0)^2\right\}}{2\sqrt{2\pi n\rho s k}}=5.5 \text{ deg. C.}$$

corresponding to a total surface fluctuation of 11 degrees C. For a cast-iron piston $\rho = 7.8$, $s = 0.11$, $k = 0.11$, and adopting the same values for e, T_1 and T_2, but assuming $\theta_0 = 400$ degrees C., the fluctuation of surface temperature is 16.4 degrees C. From these figures it is apparent that at high speeds the fluctuation of surface temperature is, for practical purposes, negligible.

Heat Inflow to Piston.—Taking the heat inflow to the piston as equal to "h" in C.G.S. units, the heat inflow in C.H.U. per square foot per minute will be—

$$\frac{h\times 60\times 930}{453.6}=123\,h.$$

Thus, in the author's 100 × 140 millimeter aluminum pistons, the mean heat inflow over the whole area of the piston is 58.9 C.H.U. per minute, or the heat equivalent of 2.50 H.P. The average brake horsepower in the trials (Table I) was 18.0, so that the heat given to the piston in this case is 0.139 of the heat equivalent to the brake horsepower. The heat supplied to the engine per minute = 1675 C.H.U., so that the heat given to the piston has a mean value of 3.5 per cent of the heat of combustion of the fuel.

Corresponding calculations from data given in tests of gas engines by Hopkinson,§ Burstall,‖ and Coker,¶ show the following mean results:—

Authority.	Heat to Piston, as a Percentage of,		
	B.H.P.	Heat of combustion of Fuel.	Total Heat Loss to surface of Piston and Walls.
Hopkinson ..	15·6*	3·6	12·2
Burstall ..	11·8*	3·7	10·3
Coker ..	16·2*	4·6	13·1
Author ..	13·9	3·5	15·0

The tests by Hopkinson were carried out on an engine 11.5 inches × 21 inches, with a compression ratio of 6.37, at 180 R.P.M.; those by Coker on an engine 7 inches × 15 inches with a ratio of 5.8, at 200 R.P.M.; and those by Burstall on a 16 inch × 24 inch engine, using producer gas and having a water-cooled piston. In these tests the compression ratio was varied from 4.36 to 8.07. The proportion of heat given to the piston varied in an irregular manner with the compression ratio, between the limits 3.03 per cent and 4.95 per cent, having a mean value of 3.7 per cent. On the whole, the results tend to indicate that in these engines, although of such widely different types and sizes, sensibly the same proportion of the heat of combustion is transmitted to the piston. The difference between the figures given in the last column of the above table is largely explained

* In these tests the I.H.P. was measured. It has been assumed that the mechanical efficiency was 85 per cent in deducing corresponding values for the B.H.P.

§ Proc. Inst. C. E., vol. clxxvi, Part II.

‖ Proc. Inst. C. E., 1908, Part I, page 5.

¶ Proc. Inst. C. E., vol. cxcvi, Part II.

by the fact that the ratio of the piston area to the total exposed area is greater in the author's engine and in that of Burstall, than in the others. Comparing the small high-speed engine with that of Hopkinson, when the piston is fully out this ratio is 17 per cent greater, so that, had the stroke-bore ratio been the same in both, the figure for the latter tests would be approximately 14.3 per cent, as compared with the author's value of 15 per cent. The corresponding figures for the tests of Burstall and Coker would be respectively 16.4 and 17.7 per cent.

The Burning of Aluminium Pistons.—In the course of the experiments many cases of the burning of aluminium pistons have been brought before the author's notice. These present features of considerable interest, in that the burning invariably takes place at the periphery and not near the center, where, under normal conditions, the maximum temperatures would be expected to occur. Moreover, in no case in which piston temperatures have been measured in operation, even in cylinders in which pistons have previously given trouble, has the maximum measured temperature been nearly high enough to cause trouble of this kind. The investigation of the probable cyclical fluctuations of temperature in the surface of the metal indicates that this is not sufficiently great to affect the results, although in this connection it should be pointed out that in this investigation no account is taken of the possibility of the material being porous. If appreciably porous, it is conceivable that the hot gases being driven into the surface porosities on the explosion stroke might produce local cyclical variations of magnitude much greater than those calculated on the assumption that this factor is negligible. On the other hand, it would be expected that any such porosities would be rapidly filled by oil carbonizing on the top of the piston, while even if not, burning would still be expected to take place near the piston crown rather than at its edge,

The following facts have been noticed in connection with these burnt pistons.

(1) A burnt piston is always one in which the clearance between the cylinder-wall and the piston over the top lands, is excessive.

(2) While failures are not confined to any one quadrant of the piston-head, with only one sparking-plug the great majority of failures take place at a point on the diameter opposite the plug, while with two plugs they take place on the diameter joining the plugs, sometimes under one and sometimes under the other plug.

(3) The tendency to burn appears to be greatest in those cylinders fitted with inclined sparking-plugs.

Excessive clearance is obviously to be deprecated as admitting the hot gases between the piston crown and the cylinder walls. This increases the temperature of the piston crown, both directly and indirectly, in that it increases the temperature of the cylinder wall. From general considerations the effect of the sparking-plug positions would be expected to be much as indicated, since, other things being equal, the hottest points will be found either underneath or opposite the plugs. While all these factors contribute to the burning of a piston, it is probable, however, that they are only effective in so far as they increase the local temperature either of piston or cylinder wall, or both, to a point at which local breakdown of lubrication is produced, after which the friction of the piston against the wall produces a temperature sufficiently high to produce burning. While this is not conclusively proven, it is suggestive that most cases of burning have taken place on engines in which the supply of lubricant to the cylinder was below the normal, while in one engine, in which no trouble had been previously experienced, burning took place during lubrication experiments in which a new system, giving a reduced supply to the cylinders, was used.—"Engineering Journal," 29 January, 1926.

A 15,000 B.H.P. OIL ENGINE.

Although the engine illustrated is not intended for marine work, it is of interest as showing that oil engines can be built of larger powers than any that have hitherto been installed in ships. The unit in question is designed to develop 15,000 B.H.P. at 94 R.P.M., and has been constructed by Blohm and Voss under license from the Maschinenfabrik Augsburg Nürnberg. In general design it follows the lines of the engine fitted in the *Magdeburg*. It has nine cylinders, with a bore of 860 millimeters and a stroke of 1500 millimeters. The overall length is 23.4 meters, or about 77 feet, whilst the maximum width is 4.3 meters, or about 14 feet. The height to the top of the valves from the center of the shaft is about 33 feet 6 inches, and from the bottom of the oil sump some 38 feet 6 inches.

The engine drives two injection-air compressors, but the scavenging air is supplied from separately driven turbo blowers. The valves both at the top of the cylinders and at the bottom are all operated from one camshaft, and port scavenging is employed, according to the special M.A.N. system. This big Diesel motor is to be installed in the Hamburg Electricity Works, coupled to an alternator. The installation is expected to be completed early this year, after which prolonged trials will be carried out. In the builders' works, owing to the size of the engine, only preliminary tests have been made, apart from full-load trials on one cylinder.—"British Motor Ship," February, 1926.

EFFECT OF USE UPON THE STEAM RATE OF TURBINE-GENERATORS.

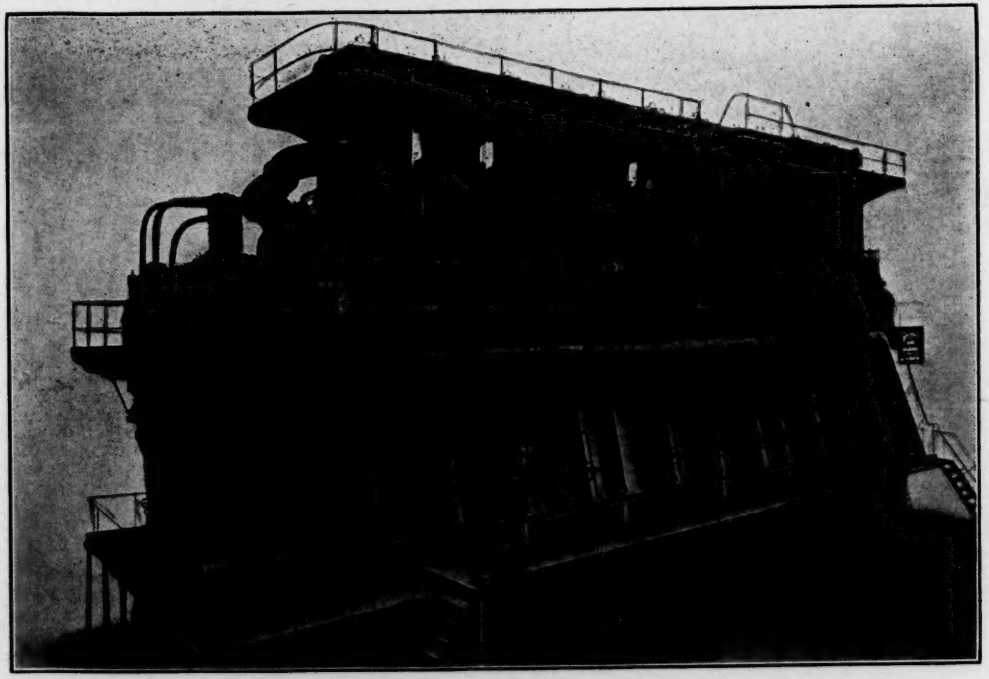
By Sterling S. Sanford.*

An interesting study has recently been made of the life history of a number of steam turbines installed in several central-station power plants to determine the effect of use and condition upon their steam rates. Several of the machines were installed previous to 1914 and one had been in service since 1909. Of the number studied, ten machines, ranging in capacity from 8000 to 20,000 K.W., had been in service and in good condition long enough to give a reliable indication of the increase in steam rate with use of the machine.

The average annual increase in steam rate of these ten machines was 0.16 pound of steam per kilowatt-hour over an average period of 5.7 years. This corresponds to an average annual increase of 1.2 per cent of the guaranteed steam rate. The average annual increase in steam rate for the individual machines ranged from 0.11 to 0.25 pound per kilowatt-hour. These values were encountered in properly maintained power plants and apply to machines in normal operating service. All the turbines were of the Curtis type, two of them being horizontal and the remainder being vertical machines. The average increase in steam rate for the horizontal units was slightly lower than for the vertical units.

Figure 1 shows graphically the effect of use upon the steam rate of one of these turbine-generators. This figure shows the steam-rate curves for various tests on a 14-000-K.W. vertical unit from December, 1911, to July, 1921. The tests are numbered in chronological order, and each curve is labeled with the date of the test. In each case the steam rate as determined by test has been corrected to standard conditions of steam pressure and temperature at the turbine throttle and pressure at the exhaust by using

* The Detroit Edison Co., Detroit, Mich.



THE LARGEST OIL ENGINE IN THE WORLD.

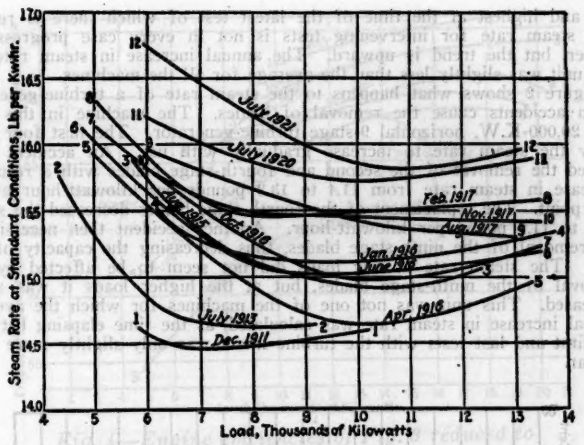


Fig. 1—Steam rate of a 14,000-kw. vertical turbine-generator over a period of ten years

the correction factors furnished by the manufacturer. The standard conditions used were those for which the guarantee was made. The effect of variation in conditions at the throttle and exhaust of the turbine thus eliminated, the variation in steam rate for different tests must be explained by changes in the internal condition of the turbine. It will be noted from Figure 1 that the steam rate was lowest at the time of the first recorded

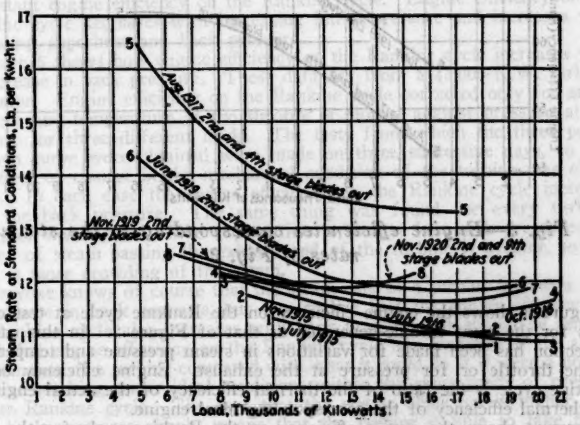

Steam Rate at Standard Conditions, Lb. per Kw-hr.
Load, Thousands of Kilowatts
Aug. 1917 2nd and 4th stage blades out
June 1919 2nd stage blades out
Nov. 1919 2nd stage blades out
Nov. 1920 2nd and 9th stage blades out
Nov. 1915
July 1915
July 1916
Oct. 1916

Fig. 2—Steam rate of a 20,000-kw. horizontal machine with various bucket wheels removed

test and highest at the time of the latest test of which there is record. The steam rate for intervening tests is not in every case progressively higher, but the trend is upward. The annual increase in steam rate for this unit was slightly less than the average for all the machines.

Figure 2 shows what happens to the steam rate of a turbine-generator when accidents cause the removal of blades. The machine in this case is a 20,000-K.W. horizontal 9-stage turbine-generator. The first four tests show the steam rate to increase gradually with use. An accident then caused the removal of the second and fourth-stage blades with a resulting increase in steam rate from 11.4 to 13.3 pounds per kilowatt-hour at the best point. The replacement of the fourth-stage blades decreased the steam rate to 11.7 pound per kilowatt-hour. Another accident then necessitated the removal of the ninth-stage blades, thus decreasing the capacity of the unit. The steam rate at low loads did not seem to be affected by the removal of the ninth-stage blades, but at the higher loads it was again increased. This unit was not one of the machines for which the average annual increase in steam rate was calculated, as the time elapsing between the first and last tests with the turbine intact was only slightly more than a year.

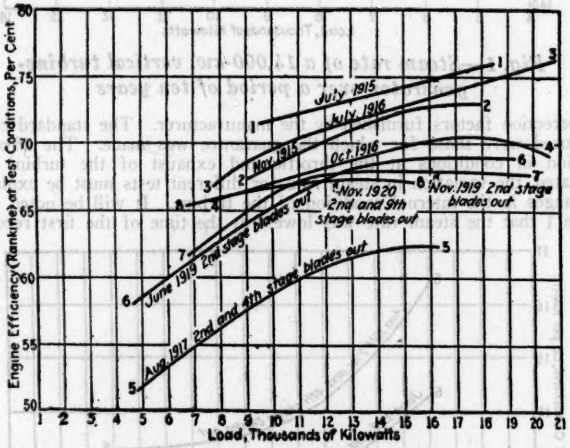

Engine Efficiency (Rankine) or Test Conditions, Per Cent
80
75
70
65
60
55
50
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21
Load, Thousands of Kilowatts
July 1915
July 1916
Nov. 1915
Oct. 1916
June 1919 2nd stage blades out
Aug. 1917 2nd and 4th Stage blades out
Nov. 1920 2nd and 9th Stage blades out
Nov. 1919 2nd stage blades out

Fig. 3—Engine efficiencies corresponding to the steam rates of Fig. 2

Figure 3 shows the engine efficiency on the Rankine cycle at test conditions for the same turbine-generator as that of Figure 2. In this case no correction has been made for variations in steam pressure and temperature at the throttle or for pressure at the exhaust. Engine efficiency on the Rankine cycle is the ratio of the thermal efficiency of the actual engine to the thermal efficiency of the corresponding ideal engine.

Figure 4 shows the engine efficiency on the Rankine cycle for the same turbine-generator at standard conditions. In this case the steam rate was corrected for variations in conditions at the throttle and exhaust before

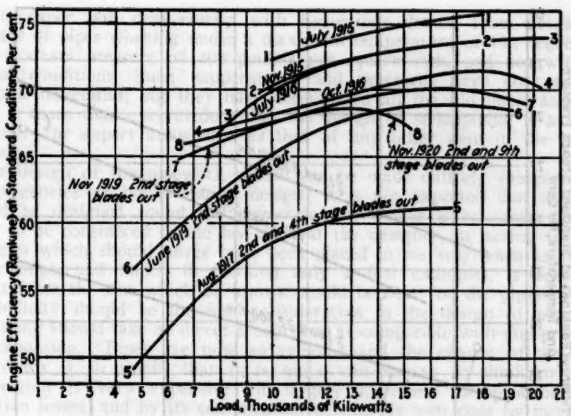

Engine Efficiency (Rankine) at Standard Conditions, Per Cent
75
70
65
60
55
50
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21
Load, Thousands of Kilowatts
July 1915
Nov. 1915
July 1918
Oct. 1916
Nov. 1919 2nd stage blades out
June 1919 2nd Stage blades out
Aug. 1917 2nd and 4th stage blades out
Nov. 1920 2nd and 9th stage blades out

Fig. 4—Engine efficiencies of Fig. 3 reduced to standard conditions

using it in computing the engine efficiency. A comparison of Figures 3 and 4 shows how engine efficiency varies with steam pressure and temperature and back pressure. Curves 2 and 3 have changed places, and curves 6 and 7 have also changed their relative positions. This demonstrates that it is not possible to correct the steam rate of a turbine-generator for variations in steam pressure and temperature and back pressure by assuming a constant engine efficiency on the Rankine cycle. Engine efficiency on the Rankine cycle decreases with increasing initial pressure and increases with increasing superheat and back pressure.

Figure 5 shows how engine efficiency on the Rankine cycle increases with an increase in back pressure. These data are from a 45,000-K.W. turbine-generator. Engine efficiency on the Rankine cycle corrected only for steam pressure and temperature at the throttle is plotted against pressure at the exhaust for three different loads. The tests from which the three points on each curve were obtained were made on three successive days, so that the effect of change in the mechanical condition of the turbine was eliminated. In each case the engine efficiency on the Rankine cycle increases with the back pressure. The same thing was found for every turbine-generator investigated. The explanation is that at low back pressures the volume of steam passing the exhaust end of the turbine is greater, so that there is more crowding of the steam.

Everyone knows of course that the steam rate of a turbine increases with an increase in back pressure, but few realize that the engine efficiency on the Rankine cycle also increases with back pressure. It should be remembered that engine efficiency on the Rankine cycle is a measure of the actual performance of a turbine compared with what it could do if it were a perfect machine working under ideal conditions within the same limits of pressure on the Rankine cycle. The fact that the engine efficiency is higher with high back pressures simply means that the turbine is able to take better advantage of the limits of pressure imposed upon it than it is at low back pressures.

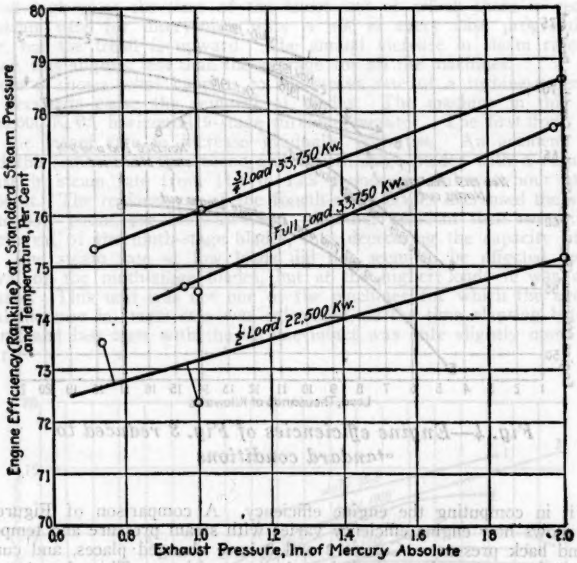


Fig. 5—Effect of exhaust pressure on the engine efficiency of a 45,000-kw. unit

It is well known that turbine buckets and nozzles deteriorate with use, notably through erosion by wet steam in the low-pressure stages, through the accumulation of scale and through accidental mechanical damage. Unfortunately, there is available little information whereby the influence of such deterioration may be accurately appraised in terms of economic performance. The data presented in this article are too scanty to support general conclusions, but are presented simply as a contribution to the information on this subject.—"Power," March 23, 1926.

STEAMPIPES FOR EXTRA HIGH PRESSURE AND TEMPERATURE.*

By J. Arthur Aiton.

During the last few years, and under the stress of competition from other types of prime movers, steam pressures have steadily risen, and temperatures are commonly in use which are about as high as possible with present materials. Any further serious rise in temperature will necessitate reconsideration of the material of which steampipes can be made, but this is not yet a live question. Increase of pressure does not necessitate any radical change in the pipes, as it can be met by increased thickness of wall, and the tendency with increased pressure is towards reduced diameter of pipes.

* Paper read before the Institution of Engineers and Shipbuilders in Scotland.

This paper will deal chiefly with deductions drawn from actual experience of pipes working under a maximum temperature of 750 degrees F., with a steam pressure of 400 pounds per square inch, and not with abnormal conditions. Such temperatures and pressures have not produced any new difficulties, but they have accentuated all the old ones, and have made it more than ever evident that the design of steampiping is as much a matter for expert knowledge as that of any other item in the power-station.

The design of the pipework should not be made entirely subservient to the exigencies of power-station design; it is not expected that a station should be designed round the pipework, but piping is so often the last thing to be considered in the lay-out that the designer, in having to avoid obstacles which should never have been placed in his way, such as pumps, fan casings, and tanks, to mention only a few examples, is forced to depart from the run which he knows would be best for the pipes to take.

Simplicity should be the first consideration in the design of pipework, and piping should take as direct a course as is compatible with due provision for expansion. Pipes are now so reliable, and the chance of an entire breakdown is so small, that it is quite unnecessary to duplicate them. Duplication is very expensive; it largely increases the probability of radiation losses, and by its complication defeats its own ends by increasing the risk of breakdown. The diameter of the pipes should be kept small by a high-velocity flow which is obtainable by a drop in pressure, say as much as 10 pounds. This may sound high, but is not serious when compared with the initial pressure, and any loss which arises from this drop is fully compensated for by reduced radiation loss and by a reduction in capital charge. It may be remembered that full drop is incurred only at maximum output, and falls very quickly, at lower loads.

Having settled the size of the pipes and generally how they are to run, the next problem to be tackled is that of expansion. With high superheat this is indeed a serious question, and is the ultimate cause of most of the trouble in pipelines. Even troubles with joints may often be traced back to undue stress on the flanges by expansion of the pipes.

It is convenient to consider expansion under two classes. First, the natural, which is due to the variations from dead cold to the highest steam temperatures and cannot be avoided, as it is inherent in the use of high superheat. Secondly, that due to extraneous causes, which may be to some extent inherent, but can be largely, if not entirely, eliminated. It is important that this distinction should be remembered, for in the first case the design will have to provide sufficient flexibility to take up the movement due to expansion, whereas in the other the design will aim at removing the cause, and so the movement. The first case can be easily met by providing wide bends in the run of the pipes. This is the most desirable arrangement, but where it is not possible the next best thing is to introduce expansion bends of the lyre type. It is advisable to allow for the pipes to be drawn up when cold to the amount of about one-third of the expansion arranged for. It may be accepted that sliding expansion pieces, however alluring they may look on paper, are not suitable for the conditions being dealt with here.

In order to be effective, expansion bends must be large, and consequently they take up considerable space, which is often difficult to secure. To surmount this difficulty, bends made from corrugated pipes are now being used, as they have much greater flexibility than those made from plain pipes. It is, therefore, possible to make them of much smaller dimensions and still obtain the same amount of movement for expansion; or, conversely, if made to the same dimensions, five times the amount of movement for expansion can be obtained, and, moreover, much smaller thrust is required in order to spring them. Corrugated pipes are stronger

than plain pipes, because the process of manufacture thickens the crown and the base of the corrugations, and they can only be made from the best quality of tube. They can be bent very easily, and the smaller sizes cold, but as this punishes the metal it is not recommended as a regular practice, although it may be a very useful feature in fitting bends into position. Experience shows that it is better to heat the pipes before bending, but it is not necessary to pack them with sand.

Corrugated pipes can be bent to a much smaller radius than plain pipes, even small enough to enable them to take the place of a cast elbow. There is very little expansion movement with such a short bend, but it enables wrought steel to take the place of a casting, and will often eliminate a pair of flanges. In ease of bending, corrugated wrought-steel tubes compete favorably with plain copper tubes, and are twice as flexible. As final bends to turbines their added flexibility in all directions is a great advantage, for they not only give more movement along the axis of the bend, but they also give considerable movement transversely without engendering a heavy thrust on the turbines. The friction loss in steam passing round corrugated bends is double that in bends made of plain pipe of the same radius, but as the number of bends used can be reduced, in favorable circumstances, to one-fifth, there is a large margin of gain in this respect. If it is impossible to reduce the number of bends, the difficulty can be overcome by fitting a large-bore pipe, which would not necessitate a bend of larger radius, nor would it curtail the movement available for expansion.

A method of making a bend from plain pipe to a radius much smaller than that to which plain tube can be bent is by cutting pieces out of the pipe, bending it with the cuts on the inside of the bend until the sides of each cut meet, and then welding up the cuts. This is a method which should not be used with steampipes for any great pressure, and certainly not with the pipes under consideration.

There are other classes of expansion which can be at least minimized by care in design. Water in pipes is the cause of the most serious stresses. It is not possible entirely to prevent it from getting into the pipes, but the amount which passes along with the steam does not do much harm. Water, however, which is allowed to collect and lodge in the pipes may do considerable damage. When steam is turned into horizontal pipes in which water has been allowed to collect, unequal expansion is set up in the pipes, hogging takes place, and stresses arise the ultimate effect of which cannot be calculated.

All pockets where water may lodge must be avoided at all costs. Valves are a great cause of water pockets, and if no valves were required in a pipe-line it would be possible to keep water from lodging in the pipes by making all ranges drain into suitable headers, but although this is not possible it would be kept in mind, and the number of valves reduced to a bare minimum. Simplicity of pipe design is a great desideratum, for redundant valves are a frequent cause of water trouble. Again, valves are often placed in positions where they are bound to form water pockets, as at the bottom of drop legs, but this cannot be wholly prevented, and when valves have to be placed in such a position they should be by-passed for drainage. Valves, of course, are not the only cause of water pockets. Bends rise off valves, causing a pocket above the valve, and expansion bends are set in a vertical position in mains, possibly causing two pockets. These are very obvious cases, but experience shows that constant watchfulness is necessary to prevent such errors creeping into otherwise well-designed pipe-lines and causing trouble in working.

Injudicious anchoring causes expansion stresses by endeavoring to force the pipe into a position where it would not naturally go. The ideal plan would be to have no anchoring at all, as under such circumstances it would

only be necessary to guide the pipe in the direction in which expansion should take place, but this ideal can seldom if ever be obtained, although it should be aimed at in all cases.

The working of modern steampipe lines is a new problem, and must be treated in a new way, for that which can be done with comparative impunity on lower-temperature work will cause very serious trouble if attempted with high-temperature work. For instance, to bring a boiler on to a main carrying superheated steam ought to be a very slow process if trouble with pipes is to be avoided.

Referring to questions of the design and manufacture of the pipes themselves, undoubtedly the weakest part of the pipe-line is the joint, and as every joint is a potential cause of trouble, the obvious aim should be to reduce the number of joints to a bare minimum by making each pipe as long as possible. As castings, even of steel, are no longer used in first-class pipework, and as modern methods allow branches to be fixed to any part of the pipe, it is possible to use straight pipes up to 35 feet or even more, and 30 feet is quite a common length. Obviously it is not always possible to take advantage of the longest lengths of tube, especially with bends, owing to the lack of facilities for transport.

Elimination of joints by welding up pipes in long lengths on the site is very attractive, but there has not yet been sufficient experience to justify recommending its adoption for the highest pressures. At any rate, great care must be taken to prevent cross stresses due to expansion coming upon the joints, or, if this cannot be done the joint will have to be reinforced to resist such stresses. A joint which aims at this result is that known as the "Sargol" joint, used in America, in which the end of the pipe is rolled over and loose flanges are used. All mechanical stresses are taken by the flanges, and a steam-tight joint is made by welding round the edge of the rolled-up portion of the pipe. This system has its drawbacks which have so far prevented its general adoption in this country, where fixed flanges are almost universal for high-pressure work.

In the use of flanges for piping there are two main questions to consider—namely, how to attach the flange to the pipe, and how to make the joint between the flanges. Regarding the first, there are nowadays only two methods of fastening flanges to pipes—namely, by riveting and by welding, except in the case of small pipes, which nearly always have flanges fixed by screwing. It seems on the face of it to be wrong that holes should be made in a pipe to be afterwards closed by rivets. A satisfactory job, however, can be made in this way, but if a rivet does leak under high-pressure high-temperature steam it is almost impossible to make it tight. Undoubtedly the greatest drawback is the presence of rivet heads inside the pipe, which, coming very close together, seriously reduce the area of passage. With high-velocity steam this either means a continuous loss due to a greater pressure drop, or a larger-sized pipe to pass the same weight of steam, with resultant loss in working due to increased radiation surface and higher capital outlay.

There is more than one method of welding flanges to pipes. The oldest and cheapest is to heat the pipe and flange in a furnace to a welding heat, and to make the weld by hammering either by hand or by machine. To control the temperature in order to obtain a welding heat on the thick flange at the same time as on the thin pipe is by no means an easy operation, especially with the larger sizes of pipe, and the risk of failure with this method is so great that this type of welding should never be used with steam above 100-pound pressure.

The only entirely satisfactory method of fastening flanges to pipes is by welding them on by means of the carbon-arc process. A continuous weld should be formed right through the flange from back to front, and in addition a good fillet should be worked up at the back of the flanges. As this

is a process of building up the weld bit by bit, it takes considerable time, and demands skill on the part of the operator; but it should be pointed out that he is able to control the work and the temperature quite easily at every instant, provided he is given the right current, the right type of machine, and the right materials to work with.

Riveting is highly skilled work, for under existing conditions it must be perfect from the start. In the days of low-pressure saturated steam, free use of caulking would make almost any rivet tight and would cover up bad work. This is far from being the case with high-pressure conditions. In the early days of welding it was very difficult to prevent testers caulking up a pinhole leak in a weld, for a soft weld lends itself to this treatment, but highly superheated steam soon finds out the weak places, and trouble results. To attempt to caulk a leaky weld is very short-sighted policy, although it seems to be such an easy way out of the difficulty. An argument frequently used against welding flanges by any process is that so much depends upon the human element, but is this more so with welding than with most engineering work, or than with riveting?

In the past the regulation of the current, and consequently the temperature, depended upon the length of the arc maintained by the welder, a very delicate operation, and one which by the very nature of things was constantly varying with the physical condition of the man, but with specially designed plant the current remains constant within wide ranges of arc, and so the regulation of the temperature does not now depend so much upon the operator.

But accepting that there is as much possibility of trouble with riveted flanges owing to poor workmanship as with welded ones, it is contended that, however badly riveting may be done, no serious accident can happen, as the flange cannot be blown off, whereas such a thing might happen with bad welding. It certainly could happen, but except with fire-welding it is not at all probable that it would happen, and so far as I know never has happened with a carbon-arc welded flange. If the process of welding be considered, it is easy to see why this should be so. The welding is done piece by piece, and the welder goes round the flange, adding the metal in layers. By carelessness he may fail to make a weld in places, but it is highly improbable that a competent man could do his work so badly that the flange would be liable to come off and yet pass the hydraulic test, hammering under pressure, and close inspection both under pressure and after the blanks were taken off. It is not difficult to see a bad weld after the flange has been faced.

Having fastened the flange to the pipe, the next important question is how to make the joint. Many methods are employed to accomplish this, but before considering them some essentials of a good joint, whatever the method adopted, may be enumerated. The flanges must be heavy to reduce to a minimum the probability of springing, the joint must be made on a raised face inside the bolt-holes, and a ground face is very desirable, though not essential. There is no mystery about the making of high-pressure joints. By care, experience, and the use of good materials, joints made by the same methods as have been used for many years are given satisfactory service to-day—that is to say, with an ordinary corrugated metal ring and a wash of thin cement. Many such joints, subjected to the working conditions of a large power-station, have been in use for over two years, and have never been touched since they were made; but it must be added that in the original lay-out of the piping expansion stresses were reduced to a minimum, and every care taken to prevent water collecting in the pipes. Expansion stresses and water are the two great enemies of pipe joints, and water is by far the worst.

The effect of water on joints, subjected alternately to high-temperature steam and water, is to set up a chemical action with the paste used for the

joints, and so to produce a slight acidity in the water. This acidulated water seems to set up electrolytic action, which quickly destroys the joints. It has not been found possible to make a paste which will not be affected by the action of water and at the same time be satisfactory in all other respects. No good joint can be made with the wrong type of bolt. Bolts made from steel of from 30 to 35 tons tensile strength have given most satisfaction, and have been adopted as a standard. Bolts made from high-tensile steel have shown a liability to brittleness' after being subjected to variations of temperature.

Where branches have to be taken off pipes this should be done by means of welded-on stools, and experience has shown that the oxyacetylene process is the most suitable for this work. At one time branches were welded on by the carbon-arc process, but, though to all appearance the weld was quite normal, it sometimes broke down under test, and as there was a possibility that one such branch might pass the test and yet break down in working, the risk was considered too great, so this process was given up. Oxyacetylene welding is more expensive, but it has given no sign of failure in working. In branch welding there are only two thin pieces of metal to be dealt with. Therefore, the period during which the full temperature of the carbon arc can be kept on the metal with safety is very short. This is possibly the root of the trouble, and if so it is inherent in the process. The branch piece is not taken through the pipe, because it is necessary to get access to the joint from the inside both with the blowpipe and with the hammer, and as this can often only be done through the branch, the joint must be inside the branch. Cast-steel tees must be ruled out of this class of pipework, as well as riveted stools, owing to the fact that they are usually riveted by hand, are too often not satisfactory, and, as they can only be fixed near the end of the pipe, they cause a lot of otherwise unnecessary joints in the main. It is advisable to have as few joints as possible. With simple design and large units it should be possible to do away with branches, and to bring the pipes into vessels which serve as collectors and distributors as well as drainage points. Collector vessels should be placed with the axis vertical, for in this position it is easier to arrange the position of the branches and the drainage, and vessels so placed are not subject to hogging stresses which are prevalent in vessels placed horizontally when water accumulates in them.

The separation of water from steam is no new thing. A simple reversal of flow is all that is necessary, and complicated arrangements of baffles and spirals are quite out of place. There should be no joint in the water space of a separator; indeed, it is best to have no joint whatsoever in these vessels. For this reason welding is far and away the most satisfactory method of manufacture. Welded throughout, large vessels of this type are working at the highest pressures, and are giving entire satisfaction, which cannot be said of riveted vessels working alongside them and subject to exactly the same working conditions. Welded vessels have been in use for fifteen years at pressures up to 200 pounds per square inch, but when an attempt was made to apply much higher pressure to this class of vessel it was found from experiment that the methods of manufacture, although fundamentally the same, had to be modified to suit the altered condition. As a result of these modifications it may be said that vessels manufactured for high pressures have a far greater margin of safety than those formerly made for lower pressure. This illustrates the fact that no method of welding is suitable for all conditions. for the body seam is welded by the water-gas process, the ends and the flanges by the carbon-arc process, and the branches by oxyacetylene. As the walls of the vessels are thick and the body is large enough for a man to get inside, the branches are carried through the wall, and are welded both on the inside and outside.

Nothing need be said about supports, as no new problems have arisen, and drainage and covering are subjects far too important to be dealt with at the end of a paper. It may be said, however, with regard to covering that great care must be taken in the selection of suitable materials, and samples of the materials to be used should be tested both for quality and for duty. Such tests can now be carried out at the National Physical Laboratory.

I may sum up by saying that no drastic alterations are necessary either in materials or in the method of manufacture to meet modern conditions in piping, but great care and experience are necessary in all departments of the work, especially in design. An eminent engineer said lately that there were so many places in a power station where trouble could be met that he did not intend to have any with pipework; in fact, he wished to, and did, forget all about it. This desirable position can be realized, but not if the pipework is forgotten or treated as a very subordinate consideration in the design of the power-station. There is food for thought in the fact that pipework is the only important part of a station which the manufacturer is not expected to design; and, further, that in many cases he does not even come into direct contact with the engineer at all. This is not altogether right. The experience of the pipemaker should be used by the engineer, as the pipe contractor cannot be expected to guarantee the successful working of the plant or the duties to be obtained, unless he has been consulted during the design of the layout. The final word is, put more responsibility upon the pipe contractor, and exact from him as definite guarantees as those exacted from the maker of any other plant, and if competent he will gladly undertake it.—"Mechanical World," February 5, 1926.

THE TREND OF STEAM-POWER-PLANT DEVELOPMENT.

A concise statement of the progress made in the past decade, so remarkable for the developments that have taken place within it in improving the economy of steam power plants, particularly large central stations. The essential conclusion to which the author comes is that while enormous gains have been made already and further gains are still possible, other factors which have increased in relative importance, particularly increase in fixed charges, may delay any further considerable decrease in coal consumption.

Load factor has a great influence on steam-station design. The influence on costs of daily, monthly, or annual load factors may be stated briefly as follows: (*a*) Daily load factor and the form of load curve affect fuel costs; (*b*) monthly load factor affects labor and fuel costs; (*c*) annual load factor affects fixed charges.

High annual load factors will warrant considerable refinement and higher investment costs to secure increased station economy. On the other hand, extremely low annual load factors warrant only a low-cost station because fixed charges constitute a larger item in power costs while fuel costs in such cases are relatively lower. Much special equipment has been installed the use of which was not economically justified by the actual operating life of that part of the plant. The relation of load factor to plant design is now better understood, and fewer mistakes are made than in earlier designs.

Cost of power consists of fuel maintenance, operating labor, and fixed charges. Fixed charges appear to be almost double fuel costs per unit of output. Hence the foremost problem confronting designers today is the reduction of initial investment in the plant.

THEORETICAL FACTORS IN STATION DESIGN.

Design and selection of steam-station equipment are governed by two factors which have been most effective in securing high efficiency. These factors are the utilization of the available heat head, and the heat balance. The main turbines are the most efficient elements in a central station. If heat head alone were considered, then all energy transformations should take place in them and all auxiliaries should be driven by electric power from the main generators. In practice this ideal is modified by such considerations as first cost, increasing efficiency of small steam turbines, reliability against shutdown, and space requirements.

Heat balance is more comprehensive than heat-head utilization. It implies a study of the plant design as a whole so that the greatest economical amount of heat energy in the coal may be delivered as electric power to the feeders leaving the station. Due regard must be given to the cost of obtaining such performance, and all gains must be balanced against the increased cost of securing such improvements. Heat-balance studies have resulted in the use of higher steam pressures and temperatures, reheating of steam, powdered coal, air preheaters, bleeder heaters for feedwater evaporators, and the various systems of electric drives for auxiliaries.

STEAM PRESSURES AND TEMPERATURES.

Of late, rapid advances have been made in increasing working steam pressures and they have reached 1200 pounds in a boiler of conventional type in this country and 3200 pounds in an experimental boiler in England. Steam temperatures, however, have increased only to 750 degrees F. For several years the author has urged the desirability of increased steam temperature, and he demonstrated the economic possibilities of higher temperatures in a recent paper.* Increased steam temperature involves increased expense only in the superheater, the main steam header and valves, and in the first stages of the turbine. Higher boiler pressure requires increased expense for all equipment from the boiler-feed pump practically to the turbine exhaust. On the other hand, steam consumption will be decreased by either higher steam pressure or higher temperature, and smaller pumps, boilers, superheaters, piping, and condenser equipment will be required. At the present time power-plant designers look to the metallurgists to develop steels suited for higher temperatures than 750 degrees F. In the meantime the general tendency is toward higher pressures, although eventually higher temperatures than 750 degrees F. will also be used.

STRUCTURAL FEATURES OF DESIGN.

Building and foundations may account for one-fifth to one-third or more of the total cost of the plant, and they are more controllable on a given site with given requirements than steam or electrical equipment. Many variables, such as amount and topography of available ground, soil and flood conditions, circulating-water supply and levels, rail service, etc., are encountered with every power-house site. It is seldom, if ever, possible to build from the same design two power plants in different locations. Skilful arrangement of equipment is necessary to reduce building costs in any large measure.

Volume is the more generally accepted unit for the comparison of plant designs. Considering boiler plants as now built, there appear to be possibilities of further developments in design and of reduction in total volume

* Higher Pressures or Higher Temperatures, A. C. Christie and D. C. Turnbull, Jr. Report of Prime Movers Committee, N. E. L. A., on Higher Steam Pressures and Temperatures, July, 1925.

and cost of the structures. Turbine-room height and width are fixed by the requirements of the condenser and its auxiliaries, and by the clearances necessary to lift covers and remove rotors from turbines and generators. It has been suggested that vertical condensers would permit reductions in roof height and lessen turbine-room volume. Much of the electrical equipment formerly housed inside the building is now placed outside, and this

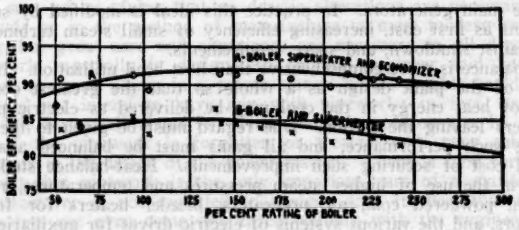


FIG. 1 TEST EFFICIENCIES OF 30,600-SQ. FT. BOILER, LAKE SHORE STATION, CLEVELAND ELECTRIC ILLUMINATING CO.

has decreased building expense. The trend now is toward simple, plain, low-cost designs, and in some cases reinforced concrete has been used in place of structural steel and brick. Much money may be needlessly spent on the station structure, and careful study of this feature can often effect large savings in total plant cost.

MODERN BOILER PLANTS.

Modern boiler plants have four outstanding characteristics: namely, increased boiler size, high ratings in operation, new furnace designs, and automatic control.

Increased boiler ratings are now possible through improved furnace designs which produce higher efficiency over a wide range of load. Combined boiler, superheater, and economizer efficiencies, as found by tests on a 30,000-square foot boiler at Lake Shore Station, are shown in Figure 1. This curve is remarkably flat over a wide range of load. Few boilers are now operated continuously at ratings above 300 per cent. Air preheaters tend to flatten the boiler-efficiency curve so that operation at peak load at much higher boiler ratings than 300 per cent does not require much sacrifice in fuel economy.

The cost of furnace and setting, air preheater, and forced and induced-draft fans will be increased when high ratings must be carried, but these increased auxiliary costs are small compared with the costs of complete additional boiler plant. High boiler ratings tend, therefore, to lower the first cost, and are often justified on this basis for the given load conditions. These higher ratings are possible only with improved furnace designs.

In this connection the author briefly discusses the relative advantages of pulverized-coal systems and stokers, and points out generally the factors which warranted the installation of the former. Powdered coal brought about important changes in furnace design, in particular, the substitution for solid walls of air-cooled brick side walls and water-cooled walls. The combination of water-cooled furnace walls and air preheaters seems in practice to offer the most economical arrangement. Future designs of boilers may be built around the furnaces rather than entirely above them.

STEAM STATIONS FOR INDUSTRIAL PLANTS.

The preceding discussion refers specifically to large central stations. Many industrial plants require a steam station to furnish steam for heating or process work or for power. The use factor of such equipment may be quite high, in which case refinements in design would be warranted that could not be justified with a low use factor.

While the same general tendencies prevail in industrial-plant design as in central stations, the former lags behind, due partly to the fact that the industrial steam plant is a secondary part of the whole works, and partly because less efficient operating management and personnel are available in the small plant than in the large electric works. Increasing boiler pressures, larger boilers of the water-tube type, pulverized coal or more modern stoker equipment, and large condensing or non-condensing steam turbines now characterize the equipment of the larger industrial plants. There appears to be a tendency in European industrial plants where process steam is required to use even higher steam pressures than in central-station practice.

Small plants are usually built with little or no thought given to fuel economy. Owners would save money by engaging competent engineering advice before building such small plants.

In the new boiler plant of a Canadian paper company, the Lopulco pulverized-coal-firing system will be installed, with a central pulverizing plant using steam driers and located near the wharf some 1200 feet away from the boilers. This plant is designed to burn any available coal, including Nova Scotia coals with low-fusing ash. An addition to the boiler room of another Canadian industrial plant consists of a 700-horsepower boiler to be initially operated at 130 pound gauge pressure but built for 350 pound gauge. A superheater to provide 100 degrees F. steam temperature will be added later. A Fuller-Bonnot unit pulverizer will furnish powdered coal to the solid-wall combustion chamber. Secondary air will be admitted through a hollow bridgewall. No water screens will be used on this plant. The unit will operate at 250 per cent rating with peaks up to 300 per cent. An old economizer now in place will be used to heat the feedwater.

The recent development of high-pressure accumulators has directed the attention of engineers to the possibilities of this equipment. In large central stations the principle of conservation of heat head, together with the flat boiler-efficiency curve over wide range of load, does not leave much opportunity for accumulators to effect savings. Another situation prevails in the process work of industrial plants, particularly where widely fluctuating steam demands are common. Heat head is then of less importance than heat capacity. [Paper presented Jan. 28, 1926, by A. G. Christie (Mem. A.S.M.E.), Professor of Mechanical Engineering, Johns Hopkins University, Baltimore, Md., at the annual meeting of the Engineering Institute of Canada, Toronto, Ont.; abstracted through "Power Plant Engineering," vol. 30, no. 4, Feb. 15, 1926, pp. 258–262, 6 figs., *gA*].—"Menchanical Engineering," April, 1926.

A NEW REFRIGERATOR.

The refrigerating apparatus which we illustrate is the invention of two young Swedish engineers, Messrs. B. de Platen and C. Munters, and is being manufactured and placed on the market by Electrolux, Ltd., of 153-155 Regent street, London, W. I. It works on the ammonia evaporation principle, and is continuous in its action. Its distinguishing feature is the fact that it embodies no moving elements whatsoever, no motor, fans, valve or other mechanical operating device forming part of its construction. For the present system embodied in the invention is being applied to refrigerating

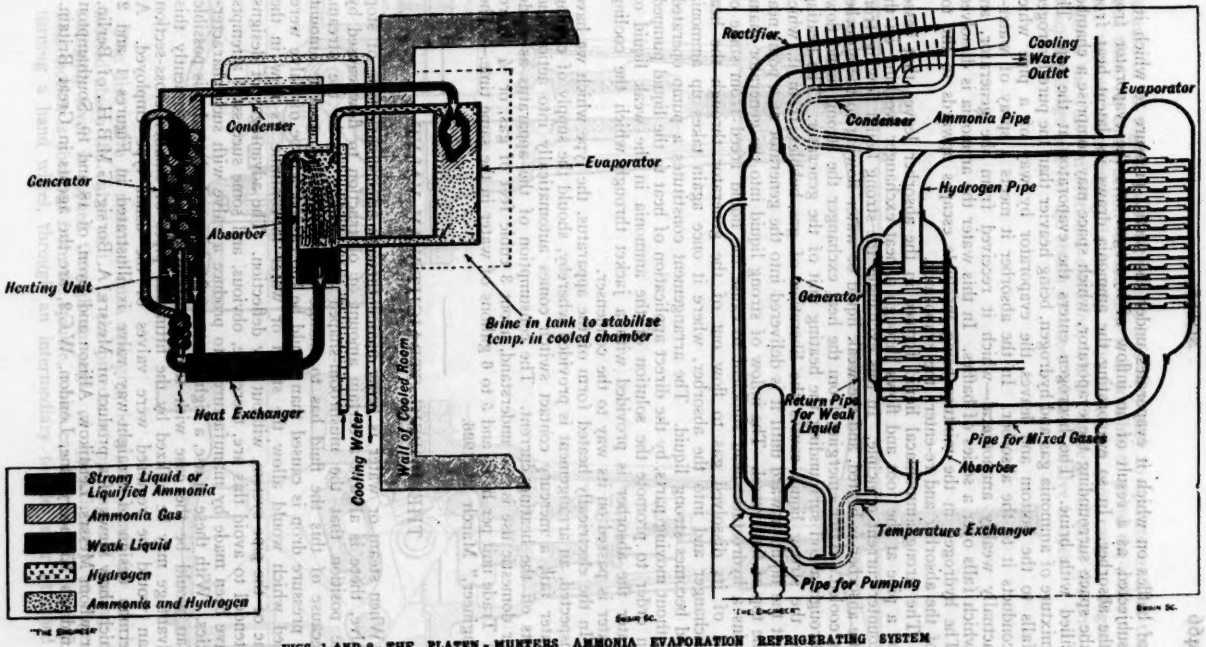
apparatus of a small size suitable for domestic and similar use, such as the preservation of food, but, we understand, its application on a larger basis is only a matter of time. The energy require to operate the apparatus is supplied either by a small gas burner, an electrical heating unit, exhaust steam or other convenient source of heat. As supplied by the makers the apparatus is charged under pressure with distilled water, ammonia and hydrogen, and as all the pipes and vessels are connected by welding, these substances remain in use an indefinitely long time. Beyond the supply of heat nothing farther is required except a circulation of cooling water.

In all continuously operating refrigerating systems there are of necessity a portion of the apparatus under relatively low pressure and another under relatively high pressure. In the ammonia evaporation system the refrigerating effect is produced by the evaporation of the ammonia in the low-pressure part. In the high-pressure portion the ammonia is condensed and the cycle is then repeated. The circulation of the working fluid between the two portions necessarily involves the expenditure of energy, which hitherto has generally been supplied by a motor or other mechanical source of power. Moreover, between the high and the low-pressure portions some form of valve has hitherto been required to effect and regulate the reduction of pressure. In the "Electrolux" refrigerator the circulation is effected on the thermosyphon principle, while the reducing valve is eliminated by an ingenious application of Dalton's law of gaseous pressures. Dalton's law, it will be recalled, states that the total pressure of a mixture of gases inside a vessel is equal to the sum of the partial pressures of the separate gases; that is to say, the sum of pressures which the gases in the mixture would exert if they were separately caused to occupy the vessel.

In the "Electrolux" refrigerator the pressure on the ammonia in the high-pressure portion of the apparatus may be of the order of 180 pounds per square inch, while in the low-pressure portion the pressure required to effect evaporation may be 30 pounds or thereabouts. The reduction of pressure is effected by introducing into the low-pressure portion hydrogen in such volume and at such pressure as to give a partial pressure of 150 pounds in the hydrogen and a partial pressure of 30 pounds in the ammonia. The hydrogen is removed from the ammonia before the ammonia circulates back into the high-pressure portion. The effective reduction of pressure on the ammonia, of 180 pounds to 30 pounds, is thus secured without the use of anything in the nature of a valve, and the total pressure throughout the apparatus is constant at 180 pounds. Other inert gases besides hydrogen might be used, but hydrogen has been chosen because its lightness greatly assists the required circulation. Not only does the introduction of the hydrogen effect the required reduction of effective pressure on the ammonia; the fact that it results in the total pressure being virtually uniform throughout the apparatus reduces the energy required to effect the circulation and indeed makes practicable the use of a simple thermo-syphon system for achieving such circulation.

The apparatus as shown in the diagram—Figure 1—and in the more complete drawing given in Figure 2, comprises three chambers, known as the generator, the absorber, and the evaporator. The evaporator is placed inside the chamber to be cooled. The generator is heated by a gas, electric or other unit and, with the absorber, is placed outside the chamber.

Inside the generator there is a mixture of ammonia and water. When this mixture is heated the ammonia is driven off and, ascending upwards, passes through a water-cooled condenser in which the gas is liquefied. Any water vapor passing off with the ammonia is condensed in an air-cooled rectifier, which, as indicated in Figure 2, is so arranged that the condensed water flows back into the generator, and does not mingle with the liquefied ammonia. Reaching the evaporator the liquefied ammonia falls on to a series



FIGS. 1 AND 2– THE PLATEN - MUNTERS AMMONIA EVAPORATION REFRIGERATING SYSTEM

of baffles on which it evaporates under the partial pressure to which it is subjected, as a result of an inflow of hydrogen into the evaporator from the absorber. In so evaporating the ammonia draws its latent heat from the space surrounding the evaporator, which space may comprise a chamber filled with brine. The hydrogen enters the evaporator at the top. The mixture of ammonia gas and hydrogen, being heavier than the pure hydrogen, falls to the bottom and leaves the evaporator by way of a pipe which conducts it to the absorber. In the absorber it meets a spray of water—actually weakly ammoniacal—which it received from the generator, and which falls over a series of baffles. In this water the ammonia is dissolved. The hydrogen in the mixture being insoluble escapes upwards to the top of the absorber and re-enters the evaporator.

The strong ammoniacal liquid formed in the absorber leaves that vessel by a pipe at the foot and flows through a heat exchanger working on the counter-current principle. In this exchanger the strong liquid flowing from the absorber is heated, and the weak liquid or water flowing to the absorber is cooled. After emerging from the heat exchanger the strong liquid flows through a coil surrounding the heating unit of the generator. The heating of the strong liquid causes the liberation of gas bubbles from it, which lift the liquid upward until it is delivered into the generator at a point near the top of that vessel. The inflow of strong liquid into the generator maintains a hydrostatic head sufficient to cause the liquid freed from some or all of its dissolved gas to flow out of the generator through the heat exchanger and into the absorber, where it once again takes up ammonia and becomes strong liquid. The arrangement constitutes a pump operated, without moving parts, by the direct application of heat to the liquid pumped. In order to promote the solution of the ammonia in the weak liquid or water the absorber is provided with a jacket through which the cooling water is passed on its way to the condenser.

In the electrically heated form of the apparatus, the type which we have inspected, an arrangement is provided whereby, should the supply of cooling water fail, a mercury contact switch comes automatically into action and cuts off the heating current. The consumption of the apparatus as made for domestic use is, we understand, about 3 cubic feet of gas, or ¼ Board of Trade unit per hour, and 5 to 6 gallons of water in the same time.—"The Engineer," March 19, 1926.

THE BORSIG STRAIGHT-WAY VALVE.

When steam or water is being passed through an ordinary type of stop valve, there is always a certain amount of obstruction to flow caused by the position that the mushroom-shaped valve takes up in the stream. Because of this the fluid has to change direction, and a greater amount of pressure drop is caused than would be the case if a type of valve were used which would allow the steam or water to continue its flow in the line of the pipe system, without any deflection. The advantages of designs intended to avoid this are, therefore, obvious, and some successful attempts have been made by manufacturers to produce a valve with such characteristics. With these in use, a higher rate of flow through the valve is possible than would be the case with the mushroom type, and consequently this advantage may be utilized by the fitting of pipes of smaller cross-section than would be required were valves of the usual type employed. A German design of straight-way valve is illustrated in Figures 1 and 2 which show views of a product of Messrs. A. Borsig, G.M.B.H., of Berlin, for whom Messrs. Leistikow, Allison and Lyon, of 18 and 19, Southampton Buildings, Chancery-lane, London, W.C.2, are the agents in Great Britain.

Because of its toggle lever movement, it is very easy to operate, and, at the same time, its construction is such that it is possible to grind the valve and seat without dismounting it in any way from the pipe range. Figure 1 shows the valve in sectional elevation, while Figure 2 indicates the unobstructed passage left for the steam or water when the valve is open.

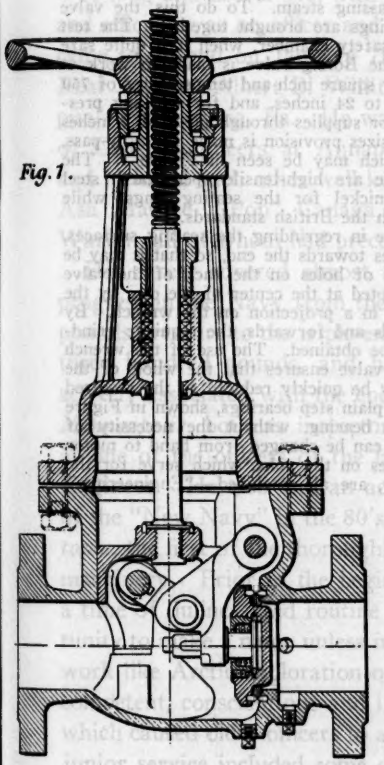


Fig. 1.

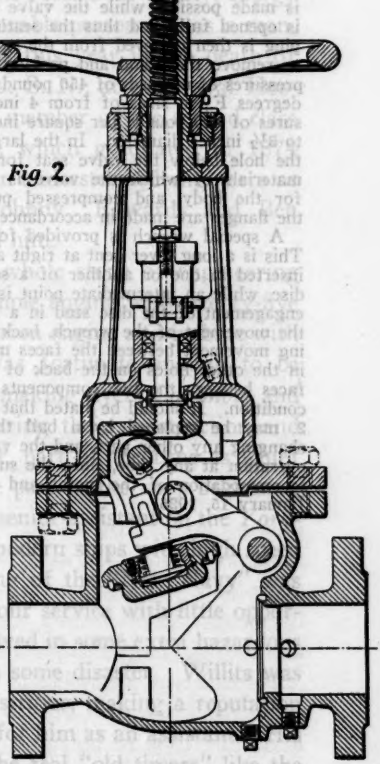


Fig. 2.

The disc valve is mounted on a swinging lever, the shaft for which is located above the valve seat. In the process of opening and closing, this swinging lever is turned through an angle of approximately 90 degrees. This motion is obtained from the spindle, which can be raised or lowered by turning a hand wheel, through an intermediary toggle lever attached to

a sliding carriage, the latter being connected with the spindle. Between the valve and the swinging lever, a spring is provided, which prevents any chatter of the valve disc on its seat. The use of the toggle lever gives a considerable mechanical advantage in opening and closing the valve, and the variation of leverage is such as to ensure that the maximum pressure of the valve on the seat is available as the valve reaches its seat and, at the time of this action, the speed of movement is a minimum.

One feature, which should be mentioned, is that when the valve is fully opened and the sliding carriage is drawn up into the cap, seating rings on the top of the carriage and on the dome of the cap shut off the safety chamber below the stuffing-box from live steam. This provision is of considerable service for, by its introduction, repacking of the stuffing-box is made possible while the valve is passing steam. To do this, the valve is opened fully and thus the seating rings are brought together. The test plug is then removed from the small safety chamber, when it is quite safe to remove the gland and packing. The Borsig valve is made to work at pressures of upwards of 450 pounds per square inch and temperatures of 750 degrees F., in sizes of from 4 inches to 24 inches, and for working pressures of 600 pounds per square inch for supplies through pipes of 2 inches to 3½ inches diameter. In the larger sizes provision is made for a by-pass, the hole below the valve seat for which may be seen in Figure 1. The materials of which the valve is made are high-tensile open-hearth steel for the body, and compressed pure nickel for the seating rings, while the flanges are made in accordance with the British standards.

A special wrench is provided for use in regrinding the seating surfaces. This is a long lever bent at right angles towards the end, so that it may be inserted in one or another of a series of holes on the back of the valve disc, while an intermediate point is pivoted at the center of the disc by the engagement of the disc stud in a hole in a projection on the wrench. By the movement of the wrench, backwards and forwards, the requisite grinding movement between the faces may be obtained. The use of the wrench in the other holes on the back of the valve ensures that the whole of the faces between the two components may be quickly reduced to the required condition. It should be stated that the plain step bearings, shown in Figure 2, may be replaced by a ball thrust bearing, without the necessity of changing any other part, and the valve can be changed from hand to motor operation at any time, since the surfaces on the yoke, which serve for the accommodation of the motor and gear, are all machined.—"Engineering," January 15, 1926.

OBITUARY.

REAR ADMIRAL ALBERT B. WILLITS, U. S. N.

A long and useful life, with a fine record for ability, energy and efficiency as an officer and a lovable personality as a man, came to an end on January 7th of this year in the death of Rear Admiral Albert B. Willits, U. S. N., at the Naval Hospital, Philadelphia, Pa. He was a member of the second class of Cadet-Engineers at Annapolis, which graduated in 1874 and included such other well known and esteemed engineers as Asa Mattice, "Pop" Canaga, "Johnny" Edwards and Ben Warren, all of whom had preceded him.

His career covered almost the whole story of engineering progress in the Navy from the simple engine with twenty-five pounds pressure to the triple-expansion engine and turbine with three hundred pounds, and he was fortunate enough to be actively associated with the epoch-making flyers *Columbia* and *Minneapolis*, both as inspector during their construction and as one of the officers of the latter, part of the time as Chief Engineer. He also had an active part in the very beginning of the "New Navy" in the 80's as senior-assistant on the *Yorktown*, the first of the thoroughly modern ships with high speed machinery. Prior to the beginning of the "New Navy" was a time of dullness and routine in our service with little opportunity to make a name unless involved in some extra hazardous work like Arctic exploration or in some disaster. Willits was competent, conscientious and industrious, making a reputation which caused older officers to ask for him as an assistant. His junior service included some of the real "old timers" like the *Hartford*, *Pensacola* and *Powhatan* and finally the *Yorktown*, *Boston*, *Minneapolis* and *Marblehead*, where he was Chief Engineer.

At the outbreak of the Spanish War he was ordered to duty in the Bureau of Steam Engineering in charge of the work at the Navy Yards. This covered the greatly increased activity due to the war and, like all his other work, was performed in a highly creditable way.

In June, 1900, Willits, then Lieutenant Commander, and Lieutenant (now Admiral) B. C. Bryan conducted a series of tests on one of the boilers for the U. S. S. *Cincinnati* at the works of The Babcock & Wilcox Co. These tests were very elaborate and secured a great deal of information which was not previously on record and which was very valuable, including such items as temperatures at various points in the tube bank for different rates of combustion. The tests included one when the coal was burned at the rate of 60 pounds per square foot of grate surface, which was the highest in a water tube boiler (other than of the Express Type) up to that time.

His next sea duty was as Chief Engineer of the *Iowa* and afterwards of the *Newark* as Fleet Engineer. He was Chief Engineer of the Norfolk Navy Yard from 1904 to 1908 and this was followed by duty at the works of the New York Shipbuilding Corporation as Senior Inspector of Machinery for the *Arkansas* and *Utah*. He was then assigned to special duty as Director of Navy Yards in 1911 and 1912, being the first officer to hold this position. He was selected for this particular duty on account of his ability and progressiveness and with a view to having the work at the Navy Yards organized in accordance with the latest experience in large industrial establishments. It involved a visit to Europe and an inspection of many of the large establishments there.

He had been promoted to be a Chief Engineer in 1896, and when the Engineer Corps was amalgamated with the Line in March of 1899 he became a Lieutenant Commander. He was promoted through the various grades and attained the rank of Rear Admiral in September, 1911. He was retired (under the then age limit of sixty-two years) in 1913.

Admiral Willits had been a member of this Society from the very beginning and during his tour of duty at the Bureau was Secretary-Treasurer and Editor of the JOURNAL for the years of 1899 and 1900. He was a clever writer and his editorial conduct of the JOURNAL fully maintained and enhanced the reputation which had been built up under the various predecessors.

He was in fine condition physically when retired, and was of too active a disposition to remain idle under such circumstances so that for a time he engaged in work as a consulting engineer with an office in Philadelphia. When our country entered the Great War, and the retired officers were recalled to duty, Admiral Willits was assigned in 1917 as Inspector of Machinery at the Works of The Babcock & Wilcox Co., Bayonne, N. J., where he continued for two years. The demands upon this Company for output during this period were very great, as they were called upon to supply some 1200 boilers for the Shipping Board in addition to several sets of battleship boilers, more than 100 Babcock & Wilcox Boilers for Mine Sweepers and some 300 Express Type Boilers, all for the Navy. This required zeal and activity on the part of the Chief Inspector, as well as his corps of assistants. All who were brought in contact with the Admiral came to regard him as the ideal inspector. He was thoroughly competent professionally and conscientious to a high degree, but at the same time he was blessed with an abundant fund of common sense which avoided those complications which occasionally arise when an inspector slavishly follows the letter of the specifications to the extinction sometimes of their spirit. The foremen and other people in the management at the Works had so much confidence in the fairness and accuracy of judgment of Admiral Willits that his attention was called by them if they had any question as to the thorough compliance of work with specifications. The presence of an inspector of such ability and fairness is a great asset to the Company where he is on duty. In effect it amounts to the Company having the benefit of the services of a very high

grade inspector without expense. His presence is a constant incentive to the very best workmanship.

Admiral Willits was a man of rare personal charm and amiability and had a host of friends in the Service as well as out of it. The people at the Works where he had been an inspector felt his death as a personal loss. He was married when only twenty-five and had a long and happy married life for forty-five years until the death of Mrs. Willits in 1921. His three daughters who survive him all married officers of the Service, Colonel Burton of the Marine Corps, Commander Crowell of the Supply Corps and Commander Calver of the Medical Corps. He is also survived by a number of grandchildren, a sister and two brothers. Funeral services were held in Philadelphia on January 8th and the interment was in the National Cemetery at Arlington on January 9th.

Admiral Willits had almost reached the age of seventy-five years, and barring occasional illness had enjoyed good health since retirement. With his sunny disposition and altruistic spirit he was passing a beautiful old age when called away. Like Abou-ben-Adhem in the poem, he could have said "Put me down as one who loves his fellow man," a feeling that was cordially reciprocated, so that his memory will long be cherished by those who had the privilege of association with him.

W. M. McFARLAND.

WALTER ALBERT SMEAD.

Commander Walter Albert Smead, U. S. Navy, died at Panama, C. Z., of heart disease, on March 10, 1926.

Commander Smead graduated from the Naval Academy in 1904. During the World War he was in charge of the Repair Division of the Bureau of Steam Engineering. So well did he handle this difficult position that he was given a letter of commendation which said in part: "As Officer in Charge of the Repair Division of the Bureau of Steam Engineering he per-

formed highly meritorious service. His energy, ability and initiative resulted in an efficient and well handled method for caring of engineering supplies."

He was executive of the U. S. S. *Wyoming* in 1919, and in 1920, as executive of the *California,* assisted in putting that vessel in commission. In 1922 he was ordered as executive of the Great Lakes Naval Training Station, and later in that year was ordered to the Navy Department. In 1925 he assumed command of the U. S. S. *Procyon,* of which vessel he was commanding officer at the time of his death.

Commander Smead had a most engaging personality which made him one of the best known and best liked officers in the Service.

JOHN A. TOBIN.

The Society regrets to announce the death of Passed Assistant Engineer John A. Tobin, U. S. Navy (retired) on January 31, 1926. Mr. Tobin was born on August 31, 1849, and entered the Navy as a second assistant engineer in 1870. He was retired in 1890. Mr. Tobin was the inventor of the Tobin bronze which has for so many years been used in marine machinery on account of its resistance to corrosion.

WILLIAM LEDYARD CATHCART.

To die in harness has been the wish of many an active worker, and it was granted in the case of Comdr. Wm. L. Cathcart, U.S.N.R.F., who passed away in his sleep on April 1st, 1926, while laid up at home with an attack of grippe, which had taken him away from his work as professor of Engineering at Webb Institute of Naval Architecture. He had spent many years of his early manhood in active duty in the old Engineer Corps of the Navy, and volunteered and was enrolled in the Service in the War with Spain in 1898 and

again in the Great World War in 1918 and 1919. All his life was filled with interest in the Navy, about which he contributed many instructive articles to the magazines and the daily press.

He was born at Mystic, Conn., August 12, 1855, so that he was in his 71st year. He entered the University of Pennsylvania in 1871, but, before completing the course there, entered the Naval Academy in 1873, graduating with honor in 1875 in the last class of the two-year course for Cadet Engineers. He performed the usual duties of a junior engineer afloat and ashore at a time of stagnation in naval affairs, but his ability had been shown and recognized so that, when Melville selected his small staff to assist him in the design of machinery for the *San Francisco,* Cathart was one of them. He has told of that work in the tribute to the late Asa M. Mattice, which was published in the JOURNAL for August, last year. It brought him in close touch with that rugged hero Melville and they became warm friends.

Having invented a clever machine for knitting, he resigned in 1891 to exploit it. Unfortunately he was not like George Westinghouse in combining the genius of the inventor with the sagacity of the man of business, and the enterprise did not succeed commercially. He accepted the Chair of Engineering in Webb's Academy and Home for Shipbuilders from 1897 to 1899 and then became Adjunct Professor of Mechanical Engineering in Columbia University 1899-1903. For a number of years he engaged in consulting work and in writing and lecturing on engineering and economic subjects and particularly on military and naval topics. He was a lucid and fluent writer with a beautiful style so that his articles carried weight, and he was recognized as an authority in matters of naval and military strategy.

In 1918 the Chair of Engineering in Webb Institute of Naval Architecture became vacant and he was persuaded because of his personal regard for the President, Mr. Stevenson Taylor, and the Vice-President, Mr. Walter M. McFarland, to take

up again his work of twenty years earlier. It was fortunate for Webb as his ability and ripe experience enabled him to greatly improve the course. Indeed, in some respects it was probably the best course in marine engineering in the country. He was an ideal instructor, a master of his subject and with a sympathetic interest in his students, who were all very fond of him.

It happened that our two foreign wars came during the times he was at Webb, but he arranged for leave and entered the Navy as a volunteer officer. In the War with Spain he was a Chief Engineer and on duty with Melville in the Bureau of Engineering. In the World War he was a Commander and on duty with Admiral Griffin in the Bureau. In both cases he rendered valuable and efficient service.

During his tour in 1898 he did valuable service for the Navy although it had to be anonymous. Appropriations had been made for several new battleships. We already had the sixteen-knot *Oregon* Class in service and two other classes under contract, but none exceeding seventeen knots, while all the rest of the world were building vessels to make around eighteen knots. It seems incredible, but it is a fact, that we were preparing for practical duplicates of the slow vessels already designed. Melville fought hard for ships to be at least as fast as those of foreign navies, but this met with actual opposition where help was most needed, and indifference elsewhere. For more than a month Cathcart prepared a series of editorials which were published in one of the great New York papers, hammering at the Department to build the faster ships. Friends of the Secretary of the Navy sent these to him daily. Finally these stirred up one of the big technical papers, which printed a two-page editorial with tables showing what every other navy was doing. This in turn stirred the Secretary, who sent the matter to the Board on Construction. As Melville had been urging the same thing, he was called on to make good with a design to drive the hulls already under consideration at eighteen knots. He did submit sketch plans of a design, but, of course,

to get the increased power, express type boilers were used. The change in distribution of machinery weights meant readjustment of others, and it was said that this could not be done in the available time. So it was decided to practically duplicate the older battleships. Fortunately, Melville was able to persuade the Secretary to call for alternative designs from the bidders. He sent to all of them the studies he had made and urged them to prepare designs that would be up to date. Fortunately, again, the shipbuilders did submit such designs and, as a result, the *Missouri* Class was built to be of as high speed as the similar vessels of other navies. They made over eighteen knots on trial. It must always have been a source of satisfaction to Cathcart that he had helped to this maintenance of efficiency.

He was the author of text books on machine design and other engineering subjects. A fine specimen of his beautiful diction is the biographical sketch of Admiral Melville in the JOURNAL for May, 1912.

He was a member of this Society from the beginning and also of the American Society of Mechanical Engineers, Society of Naval Architects and Marine Engineers and the Franklin Institute. After the retirement of Admiral Griffin he became a partner in the firm of Griffin and Cathcart, consulting engineers, with offices in New York.

W. M. MCFARLAND.

BOOK REVIEWS.

"STATIK DER BODENCONSTRUCTION DER SCHIFFE." By WALTER SCHILLING. JULIUS SPRINGER. BERLIN, 1925. 185 PAGES, 16 × 24 CM.

As the title of this book indicates, it is a highly specialized-monograph. It is, of course, a book for designers, that is, for designers who are not content to be guided by the rules of the classification societies; it deals with only a single part of the ship, the bottom; or rather with that part of the bottom which is flat; it deals with only one aspect of design of a ship's bottom, its strength under static load. The author points out that three features must be considered in figuring the strength of a ship's bottom: (*a*) the part played in longitudinal strength, (*b*) the resistance to external pressure of a panel bounded by bulkheads and sides and stiffened by a double system of supporting members, (*c*) the action of a rectangular plate supported around its periphery. His book is devoted to the second of these items. Considering the bottom as a separate member in a framed structure, the author first studies the indeterminate bending moments which act across its boundaries, and then, in detail, the action of the two systems of supporting members, the transverse floors and longitudinals. Three types of loading are considered, by uniform pressure, by concentrated weights, and by linear loads which arise in docking. For each of these the cases of three, two, one, and no longitudinals on each side are taken up.

The utility of a study like this lies only partly in the specific conclusions. The methods, which are novel in shipbuilding, might be of wide application; the extension to decks, bulk-

heads, and sides is obviously in order. So that the importance of the study is not altogether limited by the importance of the immediate problem to which the methods are applied.

It is doubtful whether the book ever will be translated. Those to whom the German habit of linking short words to form long ones is discomforting, or who have forgotten or never acquired the language will probably have to forego its nearer acquaintance. Language, however, is only one of the obstacles. I estimate that a young technical graduate who had specialized in strength of structures for three or four years and in ship work for an equal time, and who could read the book fluently, might be able to use the methods of numerical procedure which are rather fully illustrated; but to do so with certainty and benefit would require, say, six months of uninterrupted application.

Is it worth it? There are several ways in which one can react to this question. One can dismiss it with the remark: "Aw—lay it out wit' a piece o' chalk—if you can't build 'em wit'out a book you can't build 'em wit' a book." Somewhat more tenable ground is reached in the adage "cast iron is cheaper than brains." Probably there is no commercial organization in our country that would profit by paying a young technical graduate to learn to understand this book. Nevertheless as shipbuilding loses its empirical features and proceeds to more and more exactness in design, as the number of ships increases and individual vessels are more highly specialized in adaptation to specific services, the differential between success and failure becomes narrower, and what is purely academic in one age becomes practical in another. It is possible that the age in which this book may be useful is not so far away as might be supposed.

The book conforms to the high standards of German prewar typography. Perhaps it is worth saying that readers of technical German no longer have to struggle with Gothic type. That it is intended for study and not reference is shown by the lack of an index, either of topics or symbols. When "con-

stant" moment of inertia is mentioned on page 28, evidently "uniform" is intended. Our own unfamiliarity with German work is equalled by that of the author with respect to ours, as he refers on page 120 to the experiments on strength of ship girders by the Reichsmarineamt as being the only ones known to him. The width of plating acting with an attached stiffener is assumed 45 x thickness. On page 32, line 13, I find 4 as a subscript to x where exponent is intended. The use of Gothic type for hyperbolic functions beginning Equation (48a) page 43 is unfamiliar to non-Germans. Minor points, these, though doubtless the list might be extended. It is probably fair to say that a person who wants to take the next step in extension of the work in Sections 16 and 17 of Hovgaard's "Structural Design" can not do better than to work through this book in detail.

W. P. Roop.

PRACTICAL MARINE DIESEL ENGINEERING. By Louis R. Ford. Simmons-Boardman Publishing Co., 30 Church St., New York City. Price, $7.50.

With the increasing number of Diesel engine installations being made in merchant vessels, as well as in naval vessels, a knowledge of these engines becomes more and more a necessity with the sea-going engineers who will eventually be called upon to operate them. A great many books have been published on Diesel engines, which contain material of value to the designer and student. For the man in the engine room—the steam engineer who is picking up the Diesel game—there has not hitherto been a complete operator's handbook.

Mr. Ford, who is an ex-Naval Officer, has operated Diesel engines, as well as designed and developed them. His book contains an elementary treatise on the theory of the Diesel engine, the various cycles used, and the use of the indicator.

The various parts common to all engines are described, as are also practically all types of engines which have reached the commercial stage in the marine field.

There are two chapters on normal Diesel engine operation, and several chapters on derangements likely to occur and methods of adjustment to prevent such derangements. The auxiliaries employed in connection with the Diesel engine are explained and described.

This book should be of particular value to the operating engineer of motor ships, as well as to the Naval personnel attached to submarines, as the author has kept in mind at all times the point of view of the man in the engine room.

MAHOGANY—ANTIQUE AND MODERN. (E. P. DUTTON AND CO., 681 FIFTH AVE., NEW YORK CITY—PRICE $15.00.)—EDITED BY WILLIAM FARQUHAR PAYSON.

In this beautiful book, the tale of mahogany is told, from the searching out of the trees in the jungles of Central and South America and Africa, through the preparation of the lumber into the beauty of the finished product.

The account is most interestingly set forth by men who know and love their subject—Charles Over Cornelius and Francis Morris of the Metropolitan Museum of Art; Henry B. Culver, author of "The Book of Old Ships"; Ralph Erskine; Kenneth M. Murchison, A. I. A.; Meyric R. Rogers, Professor of Art, Smith College; and Karl Schmieg, master craftsman.

Over three hundred photographs are contained in the volume. Many of these are of pieces of furniture now in the American wing of the Metropolitan Museum, although many beautiful doorways and staircases are also depicted.

Of particular interest to marine engineers and naval architects is the chapter on the use of mahogany in marine architecture. The earliest known use of mahogany by civilized man was by Cortez in repairing his vessels (ante 1540).

Mahogany in the 18th Century was considered superior to oak for war vessels, as it would bury a shot without splintering. Today, mahogany is used at sea mostly for interior trim on passenger boats and yachts, and for the hulls of small power cruisers.

The Navy purchases at present more than three hundred thousand feet of mahogany each year. It is interesting to note in comparison with the use made by Cortez of mahogany, that the Naval Aircraft Factory used two hundred and fifty thousand feet in 1925.

"Mahogany" is a book not only for the interior decorator and craftsman, but also for the layman interested in beautiful things.

A. M. C.

ASSOCIATION NOTES.

The annual banquet of the Society was held at the New Willard Hotel, Washington, D. C., on March 27, 1926. An account of the dinner is given elsewhere in this issue.

At a meeting of the Council on February 2, 1926, a change in the by-laws of the Society was proposed, whereby all officers of the Navy, Coast Guard, and Marine Corps, both active and reserve, would be eligible as naval members of the Society. This change has been voted on by the Society in accordance with the by-laws and was approved of by a large majority. The new reading of the paragraphs of the by-laws involved is as follows:

Art. 14. The following shall be eligible as naval members: Commissioned and ex-commissioned officers of the regular Navy, Marine Corps and Coast Guard of the United States; warrant and ex-warrant officers of the regular Navy, Marine Corps and Coast Guard of the United States; reserve commissioned and warrant officers of the Navy and Marine Corps and temporary officers of the Coast Guard of the United States. Persons in civil life whose knowledge of engineering is such that they can cooperate with naval engineers in the promotion of professional knowledge may be eligible as Civil Members; they shall be thirty-two years of age, or over, and shall have been in the active practice of an engineering profession for at least ten years and in responsible charge of important work for five years, and shall be qualified to design as well as to direct engineering work. Fulfilling the duties of a professor of engineering who is in charge of a department in a college or school of accepted standing shall be taken as an equivalent to an equal

number of years of active practice. Graduation from a school of engineering of recognized standing shall be considered as equivalent to two years of active practice.

Art. 15. Persons eligible as naval members shall be admitted upon application and payment of annual dues. Civilians who are eligible to civil membership may be admitted upon application in writing and payment of the annual dues, provided that the application is accompanied by the recommendation of two members and provided that the application shall receive the approval of a majority of the Council.

Art. 16. Persons in civil life who are not eligible for civil membership, but who are especially interested in Naval matters or the Merchant Marine may be eligible as Associates. Commissioned officers of the United States Army and of foreign military and naval services may be eligible as Associates.

It is hoped that all members of the Society will call these changes in the by-laws to the attention of reserve officers and make a special effort to have them join the Society.

The Society is in receipt of an invitation from the Engineers Club of Philadelphia, extending the privileges of the club for a period of ten days to members of this Society who may be in Philadelphia during the Sesqui Centennial Year 1926.

A Congress of American Industry will be held in Philadelphia in September, 1926, during the Sesqui Centennial Exposition. The Congress will cover a period of three weeks (September 7 to 24). In addition to the discussion of numerous topics, there will be an exhibition of a habitat nature, portraying the habits, customs and general conditions in a manner which will visualize the industrial advancement of the United States during the one hundred and fifty-year period from 1776 to 1926.

The following members have joined the Society since the publication of the February JOURNAL:

NAVAL.

Hardy, William C., Lieutenant, U. S. N.
Kirkman, V. L., Lieutenant Commander, U. S. N.
Nuesse, Louis W., Ensign, U. S. N.
Ogden, Samuel B., Lieutenant, U. S. N.
Rule, Howard C., Lieutenant, U. S. N.
Rorschach, Frank, Jr., Lieutenant, U. S. N.
Singer, M. B., Ensign, U. S. C. G.
Tardy, Walter B., Lieutenant Commander, U. S. N. (Retired), 74 Knickerbocker Building, Baltimore, Md.
Taylor, John A., Ensign, U. S. N.
Turney, Hugh W., Lieutenant, U. S. N.

CIVIL.

Aitken, Samuel, 5 Broadway, New York, N. Y.
Buchanan, Myron W., Diehl Mfg. Co., Elizabeth, N. J.
Capstaff, James A., Capstaff & Hunter Turbine Works, 1120 Clinton St., Hoboken, N. J.
Carrier, Willis H., The Carrier Engineering Corporation, Newark, N. J.
Case, Alvin L., 177 Broad St., Portsmouth, N. H.
Cassey, Thomas E., Bureau of Engineering, Navy Department, Washington, D. C.
Ehrhart, R. N., The Elliott Co., Jeannette, Pa.
Gaskin, George Henry, 9 Page Terrace, South Orange, N. J.
Gilmour, J. B., 1522 4th St., West, Seattle, Wash.
Jansen, Johan B., Puerto Cabello, Venezuela.
McIntosh, P., Grace Lines, Inc., 10 Hanover Square, New York, N. Y.
Middleton, C. W., Babcock & Wilcox Co., 85 Liberty St., New York, N. Y.

Myers, Guy Alger, 2312 Avenue P, Galveston, Tex.
Paige, J. F., Fleet Corporation, Washington, D. C.
Pegg, Albert O., Union Oil Building, 9th and Hope Sts., Los Angeles, Calif.
Pourtless, L. S., Lykes Bros.-Rippey S. S. Co., Inc., Galveston, Tex.
Stevens, E. A., Jr., 1 Newark St., Hoboken, N. J.
Vaughan, Guy W., 238 Lewis St., Paterson, N. J.
Whitehurst, Roland, 1823 L St., N. W., Washington, D. C.
Williams, F. C., 3509 Avenue K, Brooklyn, N. Y.

ASSOCIATE.

Horn, A. F. E., General Electric Co., Washington, D. C.
MacLean, C. S., 667 77th St., Brooklyn, N. Y.
Ness, Ivar H., Winton Engine Company, Cleveland, Ohio.
Steel, John W., 2053 East 16th St., Brooklyn, N. Y.
Yates, Wm. F., 313 Machinists Building, 9th St. and Mt. Vernon Pl., N. W., Washington, D. C.